MODEL 600M
MILITARY MicroVAX II
SHOCK AND VIBRATION TEST REPORT

CODAR
TECHNOLOGY
INC.

BLANK

MODEL 600M
MILITARY MicroVAX II
SHOCK AND VIBRATION TEST REPORT

*** PRELIMINARY 1/1/87 ***

Codar Technology, Inc.
1500 Kansas Ave., Bldg. 2E
Longmont, CO 80501 USA
Telex No. 4947090
(303) 776-0472

Copyright 1987, Codar Technology, Inc.
Printed in U.S.A. 1/1/87 Revision 1.0

BLANK

TABLE OF CONTENTS

SECTION	PAGE NUMBER
SUMMARY	1
TESTING CONTRACTOR	1
TEST FACILITY	1
GOVERNMENT VERIFICATION	2
TEST FIXTURE	2
TEST CONFIGURATION	2
TEST INSTRUMENTATION	3
TEST COMPUTER	4
SYSTEM OPERATION DURING TESTING	4
TESTING RATIONAL	5
TESTING OBJECTIVE	6
TEST SEQUENCES	7
PERSONAL OBSERVATIONS	7
DISK DRIVE SHOCK TESTING	8
DISK DRIVE VIBRATION TESTING	9
PERIPHERAL SHOCK MOUNTING SYSTEM	10
BACKPLANE SHOCK & VIBRATION TESTING	11
CHASSIS SHOCK & VIBRATION TESTING	12
RESONANCE TEST RESULTS	13
TK50 SHOCK & VIBRATION TESTING	15
MIL-STD-810D BENCH HANDLING TEST	15
SYSTEM RESONANCE INTERACTIONS	16
SHOCK MOUNT SELECTION	17
REMOVABLE DISK SYSTEMS	17
SHOCK ISOLATION	18
APPENDIX A: TECHNICAL & TESTING UPDATE	
APPENDIX B: SAMPLE TEST DATA	
APPENDIX C: TECHNICAL REFERENCE PAPER	
APPENDIX D: BALL FACILITY REPORT	

BLANK

MODEL 600M SHOCK & VIBRATION TEST REPORT

SUMMARY

A production Model 600M Military MicroVAX II was submitted to Ball Aerospace Systems Division in Boulder, Colorado for MIL-STD shock and vibration testing. Testing included random vibration from 20 hz to 2000 hz at 7.9 G RMS (32 G peak); sinusoidal sweeps from 5 hz to 500 hz at 0.5, 1.5 and 4.2 Gs and multiple 11 ms shock pulses at 10, 15, 20 and 30 Gs.

Testing was performed on all three axis while a VMS software diagnostic was exercising both the CPU and a RD53 disk drive. A total of 95 test sequences were run of which 72 were plotted and documented (the remainder were pass/fail shock test). All testing was witnessed by employees of Ball Aerospace and Sperry Electronic Warfare on behalf of the Navy.

No electrical or mechanical failures occurred during testing. The 600M's disk system operated without error during the 15 G shock test. The test program demonstrated that the 600M meets the shock and vibration requirements of MIL-STDS 810C, 810D, 167-1 and MIL-E-5400T. Appendix A contains information on any additional testing which may have been completed since this report was first printed.

TESTING CONTRACTOR

All testing was conducted by Ball Aerospace Systems Division in Boulder, CO. Ball is a major defense contractor with elaborate military and NASA certified test facilities. They are equipped to handle testing on equipment up to the size of the Space Shuttle equipment bay. Testing was conducted under the supervision of Larry Uhlenkott, the facility manager.

TEST FACILITY

Ball can provide all MIL-STD-810D test sequences with the present exception of blowing sand (a rare requirement on their space pay loads). We have included a copy of their facilities brochure entitled "ENVIRONMENTAL DYNAMIC TEST CAPABILITIES AND STATIC LOAD TEST CAPABILITIES", ENV-P533. This document covers the systems which are available for shock and vibration testing.

We used the Ling Electronics B-335 vibration system for all our testing. This system is discussed on page 4 of the Ball document. The B-335 is capable of providing 18,000 force pounds.

The B-335 was driven with up to 100,000 watts under control of a GENRAD TDV-25 control and analysis system for vibration testing. Shock testing was done under control of a GENRAD 2511 system.

GOVERNMENT VERIFICATION

All testing was witnessed by engineers and QA representatives from Sperry (UNISYS) Electronic Warfare, Reston, Virginia. Sperry was representing the U.S. Navy. Testing was performed on December 18,19 & 22, 1986. Duplicates sets of test data are held by CTI, Sperry and Ball.

Sample copies of the test data are included in the appendix of this report. Due to the volume of collected data (72 pages), we have not included the entire data set. However, the complete data set is available on request.

TEST FIXTURE

The Model 600M was mounted in a special 375 pound rack mount fixture manufactured from 3" thick aluminum. This fixture was developed by a division of Martin Marietta in Denver, CO for testing of the PAWS Army MicroVAX II system. (Martin developed the PAWS MicroVAX II system, Norden is manufacturing it and CTI is designer and prime contractor for the backplane and control electronics module.)

This massive fixture bolts down to the B-335 table itself. The fixture's mass is necessary to eliminate any resonances in the fixture itself which could be attributed to the 600M by accident. An accelerometer is mounted directly to the fixture which provides feedback to the closed loop vibration and shock control system. This accelerometer also provides reference control data plots as seen throughout our data sets.

TEST CONFIGURATION

Shaker systems are only capable of operation in a single axis. Thus, it is necessary to complete all test on a given axis before proceeding to the next axis. Changing axis requires about 45 minutes between X and Y and 60 minutes to Z. Our X,Y & Z axis were defined as follows:

The X axis (horizontal) runs from front to the back of the 600M. (This axis is referred to as "longitudinal" in MIL-STD-810D.)

The Y axis (horizontal) runs from the left side to the right side. (This axis is referred to as "transverse" in MIL-STD-810D.)

The Z axis (vertical) runs from the bottom to the top of the 600M. (This axis is referred to as "vertical" in MIL-STD-810D.)

Our testing was performed in the order shown above, X,Y & Z. Both shock and vibration test were performed on a given axis before proceeding to the next axis. The X and Y horizontal testing was performed with the shaker coupled to a oil lubricated granite slip table. The test fixture with 600M is bolted to a heavy plate on the slip table. The Y axis testing is accomplished by rotating the B-335 to the vertical position and then bolting the test fixture with 600M directly to the shaker armature.

The 600M was bolted directly to the test fixture using standard production rack slides. The front of the 600M was secured to the "rack" test fixture using #10 stainless steel bolts. This configuration duplicated a standard field installation as would be used by a customer.

The rear of the box was supported only by the rack slide with one exception. The Navy likes to use rear panel "shock" pins to support the rear of a chassis. We took several sets of data in the Y axis with the 600M supported by shock pins. (The only benefit we were able to see was a reduction in acoustic noise due to a high frequency resonance in the rack slides.)

TEST INSTRUMENTATION

Four single axis accelerometers were used during testing. These were installed as follows:

1. One accelerometer was installed at the base of the test fixture. This unit provided feedback for the servo system while also providing a reference channel. This channel is identified in the data sets as "control".

2. The second accelerometer was installed inside the shock mounted peripheral enclosure on the base plate between the TK50 tape drive and a RD53 disk drive. This channel is identified as "Data 1" and "Disk drive" in the data sets.

3. The third accelerometer was mounted in the middle of a quad wide bus grant card located in the middle of the backplane. This channel is identified as "Data 2" and "Backplane" in the data sets.

4. The fourth accelerometer was mounted on the top cover of the power supply assembly and is identified in the data sets as "Data 3" and "Power supply".

The accelerometer mounting blocks were secured to the chassis components using a quick setting dental adhesive. The actual accelerometers were then screwed into the mounting blocks.

TEST COMPUTER

All testing was performed on a standard production 600M with two exceptions;

1. The test 600M was not painted to allow easier identification of any metal fatigue.

2. The machined top was removed and replaced by three aluminum angles to allow observing the interior during testing.

Initial exploratory testing on December 18 was performed using "dummy" disk drives. During this phase we were determining primarily resonant frequencies using 5 hz to 500 hz sine sweeps.

On December 19, two fully operational RD53 disk and a TK50 tape cartridge were installed in the system along with a KA630 microVAX II CPU, DHV11 and 8 Mbyte ECC memory module. This configuration was used for the duration of the test program.

SYSTEM OPERATION DURING TESTING

It was our goal to simulate real system operation during all shock and vibration testing. This was accomplished using a MicroVMS test program which performed continuous read and writes of a RD53 disk drive. (This is the same test program which was used during EMI testing). All testing was performed with the 600M running the test program, regardless of the severity of the test.

Our assumption (correctly) was that any errors would be the result of disk operations. Our program would report these errors on an external VT100 terminal. We also documented errors using the VMS system error logger. During the high level "killer" random vibration test, VMS was booted as an additional disk test.

We were concerned that the disk system might possibly "miss" the impact during the 11 Ms shock testing. We thus hit the system with between 10 and 30 shock pulses on each axis in rapid succession even though MIL-STD-810D (and C) only require 3 shock pulses on each axis.

TESTING RATIONAL

The 600M is designed for use in a variety of different military applications. Each program typically has different shock and vibration requirements. On larger programs, the 600M is usually tested to specific shock and vibration program requirements.

However, smaller programs often cannot afford special testing. As a result, we initiated a test program that would quantify the 600M's performance under a variety of different MIL-STD shock and vibration test sequences. In many cases this data will be sufficient to meet your requirements. In other cases it can be used as a starting point for additional testing.

Based on our testing experience, it would be quite easy to spend 30 days and $45,000 in rigorous shock and vibration testing. Unfortunately, it is almost impossible to find 90 days of available test time even if the funding was available. In retrospect, little useful new data would be obtained although the shear size of the data set would be rather impressive.

Our actual test program required a total of three days to complete. During this time we were able to obtain 95 shock and vibration data runs of which 72 were plotted. The remaining 23 runs were pass/fail shock testing sequences. We were able to meet all our shock/vibration test goals during this program.

At this point, the only reason for additional testing would be to quantify the performance of a 600M while mounted in a shock mounted rack or transit case. The primary goal would be to determine the resonant frequencies of the complete system, especially in tracked vehicle applications. This subject is discussed in greater detail at the end of this report.

TESTING OBJECTIVE

Our primary objective (which was successfully completed) was to pass or demonstrate the ability to pass the following MIL-STDs;

1. MIL-STD-810C, Method 516.2, Shock.
   Procedure I, Basic Design Test.
   Procedure V, Bench Handling Test.

2. MIL-STD-810D, Method 516.3, Shock.
   Procedure I, Functional Shock.
   Procedure VI, Bench Handling.

3. MIL-STD-810C, Method 514.2, Vibration.
   Procedure I, Helicopters & Aircraft.
   Procedure IA, Jet Aircraft.
   Procedure VI, Ground Vehicles.
   Procedure VII, Transportation.

4. MIL-STD-810D, Method 514.3, Vibration.
   Procedure I, Basic Transportation.
   Test Condition I-3.2.1, Vehicles.
   Test Condition I-3.2.2, Shelters.
   Test Condition I-3.2.4, Propeller Aircraft.
   Test Condition I-3.2.5, Jet Aircraft.
   Test Condition I-3.2.6, Helicopters.
   Test Condition I-3.2.10, Tracked Vehicles.
   Test Condition I-3.2.11, Ships.
   Test Condition I-3.2.12, Minimum Integrity.

5. MIL-E-5400T Airborne Electronics.
   3.2.24.5 Vibration (Jet, Propeller, Helicopter).
   3.2.24.6 Airborne Shock, 3 axes 15 G.
   3.2.24.6.2 Crash Safety Shock, 3 axes 30 G.

6. MIL-STD-167-1 Shipboard Vibration, Type I, Table 1.

MIL-STD-810D is the current revision level and should be used on all new programs. However, we still see requirements for MIL-STD-810C. The "D" revision is similar in many respects to "C". In some cases, the vibration test requirements are actually easier. This is due to the use of real world measured vibration spectra in "D" rather than the worst case "guesses" found in "C".

MIL-E-5400T vibration is similar to requirements found in 810D. The shock specification requires 18 shocks per axis at 15 Gs, 11 ms nonoperating. The crash requirement is 30 Gs, two 11 ms shock pulses per each of the 3 axes.

MIL-STD-167-1 requires a manual sinusoidal sweep from 4 hz to 50 hz with a vibratory displacement from 0.03" at 4 hz to 0.003" at 50 hz. The equipment is then vibrated for 2 hours at the resonant frequency.

TEST SEQUENCES

Based on these MIL-STDs we developed a "super set" of test which would simultaneously satisfy the requirements of numerous different test sequences and methods. Our test program for the 600M consisted of the following procedures:

1. Ten shock pulses, 20 Gs, 11 ms on each of three axis. The 600M was fully operational and running the disk test program.

2. Five shock pulses, 15 Gs, 11 ms on each of three axis. The 600M was fully operational and running the disk test program.

3. Five shock pulses, 10 Gs, 11 ms on each of three axis. The 600M was fully operational and running the disk test program.

4. Four shock pulses, 30 Gs, 11 ms on the X axis. The 600M was fully operational and running the disk test program.

5. Sine frequency sweep, 0.5 G, 5 to 500 hz, 2 octaves per minute. Four sweeps on each of 3 axes.

6. Sine frequency sweep, 1.5 G, 5 to 500 hz, 2 octaves per minute. Two sweeps on X (longitudinal) axis.

7. Sine frequency sweep, 2.5 G 8 to 33 hz, 4.2 G to 500 hz, 2 octaves/minute. Four sweeps on X axis.

8. Random vibration, 3.6 G RMS, 15 G peak, 20 to 2000 hz. Twenty minutes on each axis.

9. Random vibration, 7.9 G RMS, 31 G peak, 20 to 2000 hz. Twenty minutes on each axis.

These were minimum test requirements. In many cases the duration or number of test exceeded these requirements.

PERSONAL OBSERVATIONS

We were somewhat amazed by the severity of several of the test. The 30 G shock impulse sounds like the report from a small caliber rifle.

The 7.9 G random vibration test was deafening, even with sound deadening headsets. It was impossible to carry on a conversation within 75 feet of the test cell. The severe physical motion (vibration) has to be seen to be appreciated. It is hard to believe that a human being could survive more than 10 minutes inside a vehicle subjected to this level of vibration. By comparison, the inside of a Chinook helicopter is peaceful.

TEST RESULTS

1. The system operated without a disk error when subjected to multiple 15 G 11 ms shock pulses on all three axis.

2. At 20 Gs the system operated without causing a disk error on the X axis. It was possible, at this level, to cause disk errors about 40% of the time on the Y and Z axis.

3. All errors seemed to occur during write operations to the disk drive. Read errors were apparently automatically recovered by the disk controller.

4. The write errors seemed to be a result of the disk writing off track during the shock impulse. The disk was then unable to verify the data on the next revolution of the disk.

5. We were able to generate 30 G shock impulses only on the X axis due to limitations in the shaker system. At 30 Gs, disk errors were generated on each shock pulse. Several errors were attributed to loss of signal on the disk servo track due the severity of the shock. This resulted in a recalibration of the disk to track zero.

3. No physical or electrical damage occurred to the disk system during testing.

The disk drive was never reformatted during the three days of testing. In several cases, write errors caused VMS to lock out a copy of the actual diagnostic read/write program. When, this happened, we simply ran a different copy of the program that was stored on the same disk. We were looking for a long term degradation in the disk drive which did not seem to occur.

Following completion of testing, we ran extensive diagnostics for 72 hours at CTI on the disk drives. No problems surfaced during this test sequence or the following three weeks of general use.

Testing was performed using two Maxtor 140 Mbyte ESDI drives with a DEC MSCP emulating controller. In general, these and other disk drives are designed to operate without error with up to 2 Gs on each axis.

CONCLUSIONS

1. The Winchester disk will operate reliably without error with input shocks up to 15 Gs on all three axis.

2. Shock inputs greater than 15 Gs will result in errors during a write operation but will not damage the disk itself.

3. Maximum nonoperating shock input is estimated at 50 Gs.

TEST RESULTS

1. We were able to introduce disk errors only with the 7.9 G RMS random vibration test on the Y axis. At this level, we would see infrequent errors (1 minute intervals) which the VMS system seemed to be able to recover.

2. We were able to boot VMS from disk during the 7.9 G random vibration sequence.

3. No errors occurred with the sinusoidal sweeps or during the 3.9 G random vibration test.

4. As with shock, errors seemed to be confined to disk write operations.

CONCLUSIONS

1. The Aeroflex mounts are providing excellent isolation of the disk system as expected from the low resonant frequency.

2. Error free disk drive operation is possible under random vibration at levels up to about 6.5 Gs RMS on all axis.

3. No disk damage will occur due to sinusoidal or random vibration within the test limits in this report.

TEST RESULTS

1. The self resonant frequency of the peripheral isolation system was determined to be between 6 and 8 hz depending on axis and input driving level. This accounts for the excellent shock and vibration isolation.

2. The Q at resonance varies between 2 and 2.5. At 20 hz the Q is 0.2 (80% attenuation of the input vibration). At 100 hz the Q is 0.05.

3. Increasing the total number of Aeroflex mounts from 6 to 8 did not result in a noticeable change in the system resonant frequency.

4. At resonance (6 hz), up to about 1 G (0.4" displacement) of sinusoidal vibration can be handled in the X axis. At 1.5 Gs in the X axis, the peripheral tray hit the front door and card cage (0.6" X 2 = 1.2" displacement). The Y and Z axis can handle 1.5 Gs at resonance as the shock system is stiffer in these axis.

5. At 20 hz, the system can easily handle 5 G sinusoidal vibration.

6. The X axis is the "softest" followed by Y and Z. Shock reduction on the X axis was measured at a factor of ten. The other axis varied from three to six depending on input level. This variance is due to the difference in K (spring constant) for each of the Aeroflex's three axis.

7. Stiffer shocks will result in a higher resonant frequency. For a given acceleration the displacement is obviously less at the higher frequency. Thus, higher G levels can be handled for a given mechanical travel. However, the penalty is that less shock isolation will be provided as the resonant frequency approaches the effective shock frequency.

CONCLUSIONS

1. The present isolation system is extremely effective for both shock and vibration above about 15 hz.

2. The low frequency vibration dynamic range can be increased at the expense of shock handling capability (a standard trade off due to the laws of physics).

3. The isolation performance can be "tuned" to a specific application by using different mounts or reorienting the mounting axis of the existing mounts.

4. Actual shock and vibration to the disk HDA is actually less than shown in the data as the accelerometer was mounted on the disk tray between drives rather than on a disk drive itself.

TEST RESULTS

1. No backplane/card cage failures occurred during either the shock or vibration testing. All modules including the KA630 CPU remained fully functional through out the testing.

3. All Q-Bus modules remained seated throughout the shock testing with the one exception noted below. No external locking mechanisms were used as we were interested in determining if the boards were susceptible to self ejection. In particular, all modules remained firmly seated in spite of repeated 20 G shocks in the +Y axis.

4. We determined that a single dual wide module with an empty slot next to it will "cock" at about an 18 degree angle due to a 20+ G shock pulse in the X axis. This points out the importance of installing a blank CTI bus grant card with locking channel next to a single dual wide module.

5. The 1.5 G sinusoidal X axis vibration test resulted in a slight lifting of the ejector ears at resonance on the memory module.

6. The blank quad-wide bus grant card exhibited several high frequency resonances depending on the excitation axis. This was documented in the accelerometer data. We noticed that the resonance displacement (or Q) was highest on the bus grant card. Other modules with logic showed noticeably less displacement.

7. No module damage occurred due to vibration, either random or sinusoidal.

CONCLUSIONS

1. The backplane assembly does not require shock or vibration isolation.

2. Simple foam module retaining blocks between the backplane and lid are adequate for any conceivable level of shock or vibration.

3. No destructive or even severe self resonances were observed on the backplane modules.

4. If used, vibration isolators should have a self resonant frequency below about 25 hz.

5. All I/O cables to the backplane modules should use locking connectors.

6. All slots should be filled with modules, bus grant or other wise. Dual-wide modules should be locked together with channels.

1. No mechanical or electrical failure occurred during testing of the chassis.

2. The present rack slide mounting hardware is sufficient.

3. The standard 600M rack slides are acceptable.

4. The use of rear panel shock pins is not required.

5. No serious self resonances were discovered in the mechanical design.

6. No problems were encountered with hardware loosening under severe vibration or shock.

7. The power control module experienced no failures or resonance problems as presently mounted on top of the shock mounted disk drive assembly.

8. No failures or degradations occurred in the power supply assembly. Present production procedures were determined to be adequate.

9. The power supply assembly passes MIL-STD-810D shock and vibration with out the use of shock mounts.

10. Only one problem was encountered. Under high shock conditions, two wires pulled out of a Berg connector. It was later determined that the Berg pins were being crimped with to much wire insulation inside the pin. Some of the pins were not locking inside the connector housing. Thus, this problem was discovered during testing but was not an actual failure due to testing as the pins were loose prior to testing. This problem has since been corrected by QA.

CONCLUSIONS

1. No mechanical modifications are required to the 600M chassis.

2. Random vibration testing on production systems may be desirable as a QA tool. We would recommend about 4.0 Gs from 20 hz to 500 hz for 15 minutes per system.

RESONANCE TEST RESULTS

RESONANT FREQUENCY - GENERAL INFORMATION

Resonant frequency was determined by using a 0.5 G sine sweep from 5 hz to 500 hz on each of the three axes. For reference, the X axis is along the length of the backplane module, the Y axis is perpendicular to the board and the Z axis is vertical (the axis in which the module is inserted and removed from the backplane.

The numbers in brackets refer to the appropriate data run which is being referenced from the master data set.

RESONANT FREQUENCY - DISK DRIVE ASSEMBLY

The resonant frequency of the shock mounted disk drive assembly varied from 6 to 8 hz depending on axis and input level. The peak G force at resonance was 2.0 (Q=4) on the X axis (2), 4.0 (Q=8) on the Y axis (32) and 2.25 (Q=4.5) on the Z axis (49).

RESONANT FREQUENCY - BACKPLANE CARD

The backplane module showed two sharp resonant peaks in the X axis, one at 90 hz and a second at 225. The peak G force at the two resonances was 5 G (Q=10) (11).

In the Y axis the predominant resonance was at 65 hz with a peak G force of about 10 (Q=20) (33).

The Z axis showed multiple low level resonances from about 70 hz to 275 hz (50). The peak G level was 3.5 G (Q=7) at 275 hz.

We noticed that normal backplane modules with ICs and components showed a much lower response with a less well defined resonance. The difference between real cards and the blank bus grant card was quite noticeable to the eye.

During the three days of testing none of the backplane cards became loose or "walked" out of the backplane connector due to either sine or random vibration.

RESONANT FREQUENCY - POWER SUPPLY COVER

The power supply cover showed a X axis resonance at 220 hz with a peak G force of 4 (Q=8) (4).

The Y axis response was rather broad from 40 hz to 175 hz with a peak G force of 2 (Q=4) (34).

The Z axis showed a sharp resonance at 70 hz with a peak G force of 2.5 (Q=5). (The response at 450 hz is false and was due to the servo system dropping out due to a fixture resonance).

TK50 SHOCK & VIBRATION TESTING

The DEC TK50 cartridge tape drive in the 600M test computer was not accessed during the shock and vibration testing. The rational for this was two fold:

A. The TK50 is normally used only for loading software.

B. We chose to devote all available CPU time to exercising the disk system during the shock and vibration test sequences.

Following completion of shock and vibration testing, the 600M's TK50 was used extensively for loading diagnostics and new copies of VMS. We can thus state that the TK50 did not sustain any permanent damage during the previous shock and vibration testing. (This is probably more of a tribute to the 600M's peripheral shock mounting than to the design of the TK50 itself.)

The TK50 has developed a rather poor reputation in the industry. It now seems that many of the negative comments are actually directed at early production units. DEC seems to have cleaned up their "QA" act on the TK50. We personally have not experienced any failures or reliability problems with the TK50.

MIL-STD-810D BENCH HANDLING TEST

MIL-STD-810D, Bench Handling, requires that the 600M chassis be lifted on edge four inches and then released onto a table top. This bench handling test is performed with the system nonoperational. The test is repeated for each of the chassis four edges.

This test was performed at CTI on the 600M which had been in vibration and shock testing. As required, the 600M was operational following the test sequence. Testing was performed on January 2, 1987.

NOTE: This test procedure was repeated with the 600M executing the disk drive test program. No errors occurred.

SYSTEM RESONANCE INTERACTIONS

Many programs are now using either shock mounted racks with large Aeroflex helical coil isolators or shock isolated transit cases as manufactured by EDAC, ECS and Thermodyne. This procedure removes much of the burden for shock and vibration isolation from the rack mounted equipment itself.

Typically, the rack is designed to handle MIL-STD-901C (the infamous hammer test). This is accomplished by using Aeroflex mounts between the rack and supporting structure. Thus, the shock and vibration levels imposed on equipment within the rack is held to reasonable MIL-STD-810D rather than MIL-STD-901C levels. Transit cases will typically provide a factor of two in shock reduction.

While usually quite effective, the use of shock mounted racks can result in severe and potentially destructive double resonances. We have seen actual examples of wire Aeroflex mounts in equipment (not ours) being torn apart due to this problem.

It is imperative that the resonant frequency and "Q" of the rack system be well defined along with the resonant frequency of the actual rack mounted equipment. Otherwise, the potential results in a field environment are disastrous.

We are often asked why our backplane is not normally shock mounted even though it is available as an option. First, as shown by the test data, it is not necessary. However, the primary reason is the resonance interaction problem as discussed above.

We would like to emphasize the importance of considering the response of your complete system or installation rather than just individual components. In some ways the MIL-STD shock and vibration test sequences (810D etc.) are miss leading as they tend to focus attention on each component, such as the 600M, rather than the complete system.

We are quite willing to discuss these issues along with effective solutions in greater detail. We can also arrange for shock/vibration testing and characterization of your complete system.

SHOCK MOUNT SELECTION

As shown in this test report, the 600M has an extremely effective shock and vibration isolation system. However, it is possible to further optimize the system response and resonances toward either the low or high frequency end of the spectrum. This is accomplished primarily by changing the design and orientation of the Aeroflex mounts.

The low frequency response is a major factor in ground transportation applications in general and tracked vehicles in particular. The high frequency end is characterized primarily by high vibration environments as found in aircraft and to a certain extent, shipboard.

Regardless of application environment, it may be necessary to change the shock and vibration response to prevent interactions with your rack mounting system.

REMOVABLE DISK SYSTEMS

The majority of systems we sell use removable Winchester disk drives. These units are well protected when installed inside the 600M. However, they are susceptible to damage outside the chassis regardless of what other vendors may tell you.

In general, a Winchester is rated for about 2 Gs while operating and up to about 20 Gs nonoperating. Shock in excess of about 30 Gs will probably result in permanent damage to the drive. This level is not that difficult to achieve as witnessed by the fact that the drive manufacturers ship their drives surrounded by about 10" of foam in every direction.

We recommend that you include a shock mounted disk drive storage container in your system design. This is a necessity if you will be shipping drives to depot centers etc. CTI can provide several different types of containers or work with you on a custom design.

SHOCK ISOLATION

Assuming the available space for deflection is fixed, a shock mount can be designed for optimum attenuation of one shock level using the total available deflection. The 600M provides at least 1 " of deflection around the disk system in all three axis and is designed to reduce a 15 G 11 ms shock to 2 Gs or less per the disk drive manufacturers requirements.

Aeroflex shocks become very nonlinear on extreme compression. This attribute allows the 600M to handle up to about 50 Gs while holding the level at the disk to under 20 Gs.

Many manufacturers use rubber shock mounts. The primary problem with rubber mounts is that the K factor (spring constant) varies by about 10 between shear and compression. Thus, the shock reduction capability varies considerably depending on axis. MIL-STD-810D requires equal performance on all three axis.

Additional technical information on shock and vibration isolators can be found in appendix A.

APPENDIX A:

TECHNICAL & TESTING UPDATE

There are no additions or corrections to this technical report at this time.

BLANK

APPENDIX B:

SAMPLE TEST DATA

SHOCK PULSE TEST DATA

TEST FUNCTION : 20 G 11 msec half sine wave X axis shock pulse.

DATA RUN NUMBER 13: Control accelerometer on test fixture.

DATA RUN NUMBER 14: Disk drive response on X axis.

SINE FREQUENCY SWEEP TEST DATA

TEST FUNCTION : Sine frequency sweep from 5 hz to 500 hz at 0.5 Gs.

DATA RUN NUMBER 1: Control accelerometer on test fixture (same for X, Y & Z axis).

DATA RUN NUMBER 2: Disk drive response with X axis excitation showing resonance at 6 hz.

DATA RUN NUMBER 32: Disk drive response with Y axis excitation showing resonance at 8 hz.

DATA RUN NUMBER 49: Disk drive response with Z axis excitation showing resonance at 10 hz.

DATA RUN NUMBER 3: Quad wide back plane board showing resonance at 92 hz with X axis excitation.

X AXIS RANDOM FREQUENCY VIBRATION TEST DATA

TEST FUNCTION : Random vibration, 20 hz to 2000 hz, 7.4 Gs RMS, 30 G peak level, 810D, Method 514.3 Jet Aircraft.

DATA RUN NUMBER 25: Control accelerometer on test fixture for X axis.

DATA RUN NUMBER 26: Disk drive response with X axis excitation.

Z AXIS RANDOM FREQUENCY VIBRATION TEST DATA

TEST FUNCTION : Random vibration, 20 hz to 2000 hz, 7.37 Gs RMS, 30 G peak level, 810D, Method 514.3 Jet Aircraft.

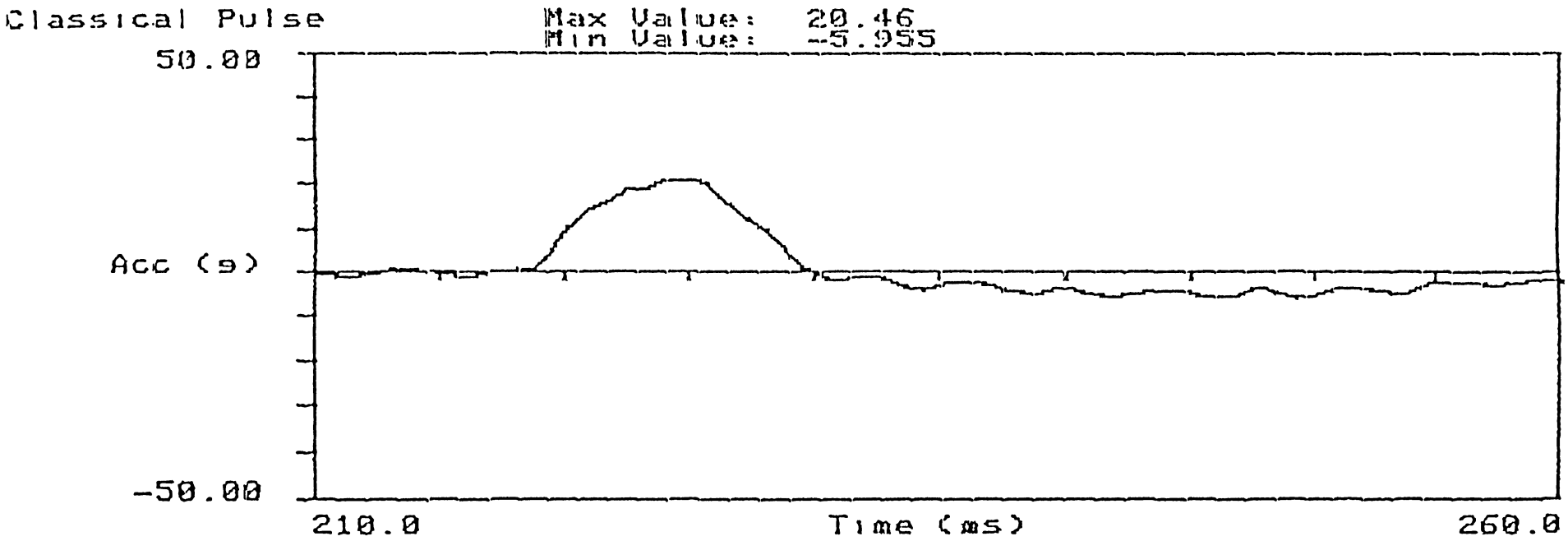
DATA RUN NUMBER 66: Control accelerometer on test fixture for Z axis.

DATA RUN NUMBER 67: Disk drive response with Z axis excitation.

DATA RUN NUMBER 68: Quad wide backplane board response (no isolation).

DATA RUN NUMBER 69: Power supply cover response (no isolation).

BLANK



18-Dec-86
13:40:10

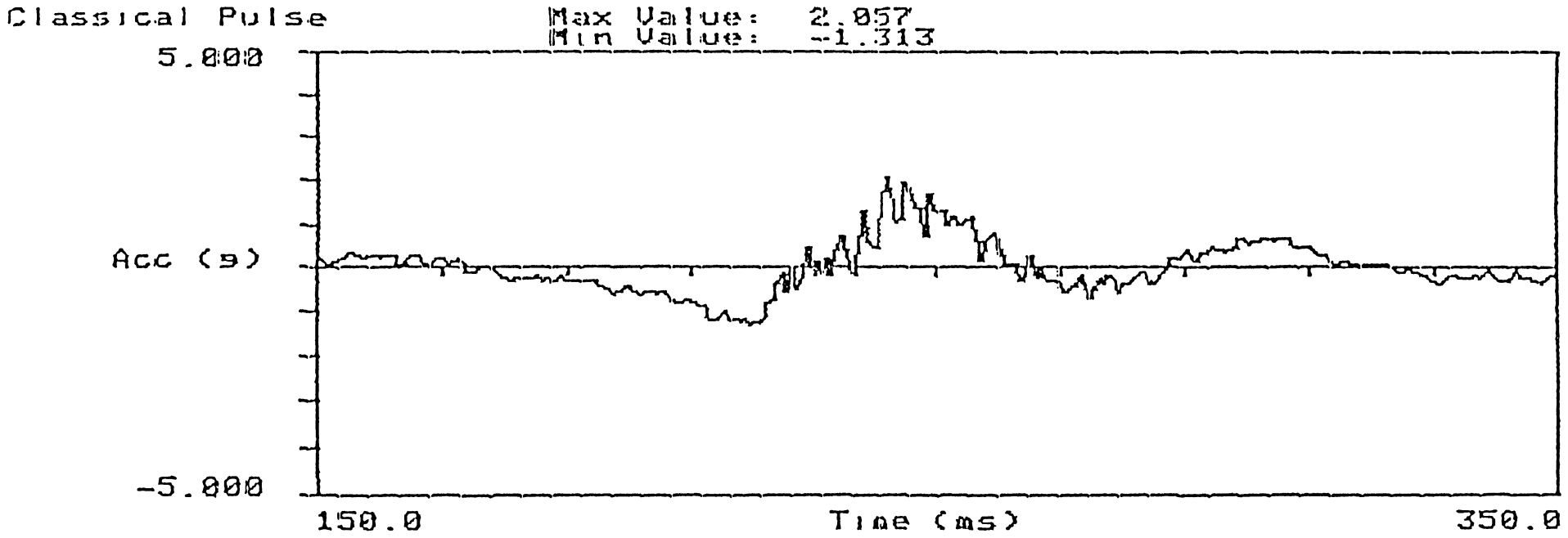
INPUT TO FIXTURE
CODAR TECH X-AXIS 20G 11MS

Level 0.0 dB

?_

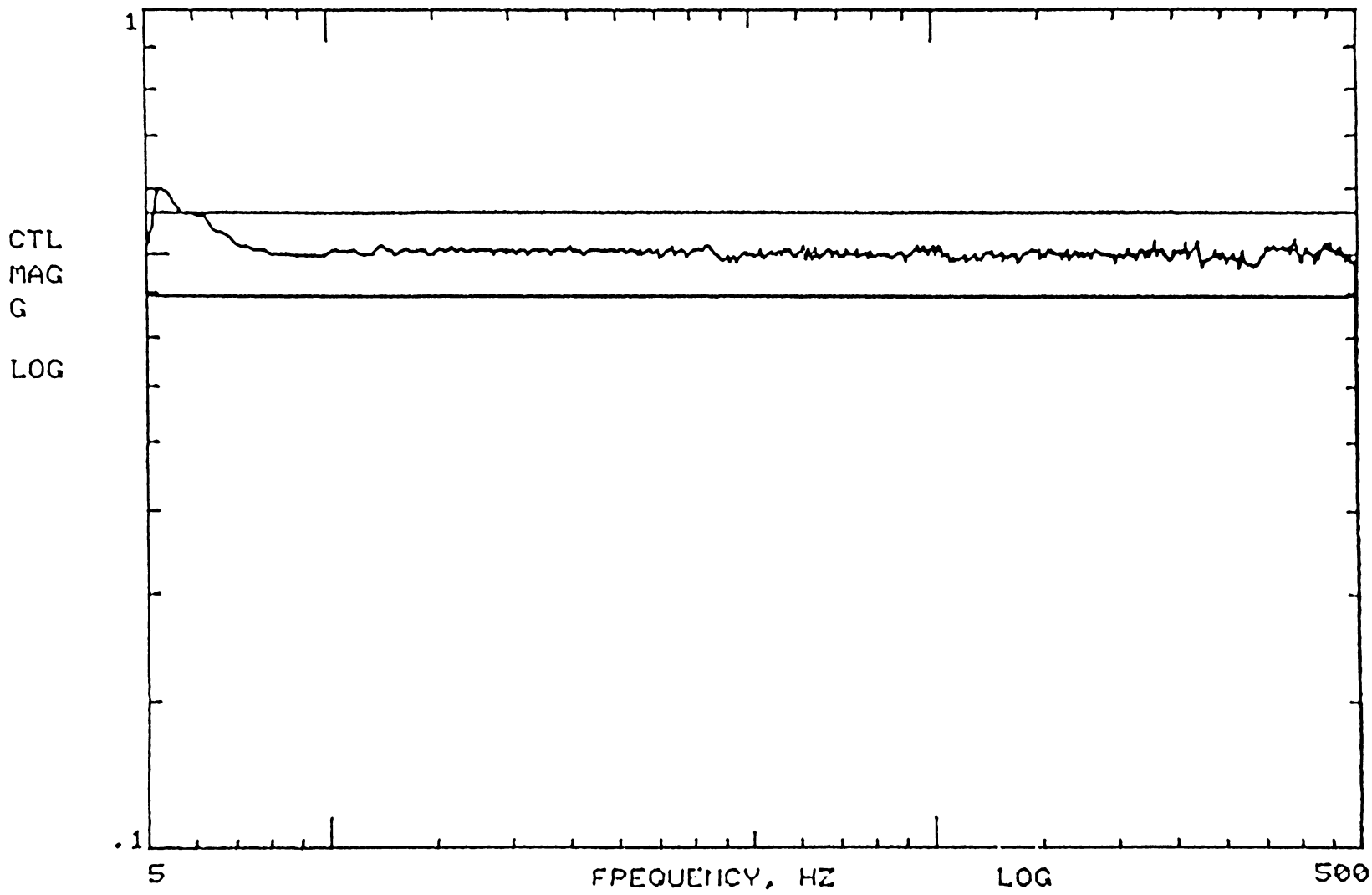
SHOCK PULSE

CODAR Technology, Inc.
600M Shock and Vibration Data
Ball Aerospace Corporation 12/86
DATA RUN Number: 13
*** Copyright (C) 1986 ***



SHOCK PULSE

CODAR Technology, Inc.
600M Shock and Vibration Data
Ball Aerospace Corporation 12/86
DATA RUN Number: 14
*** Copyright (C) 1986 ***

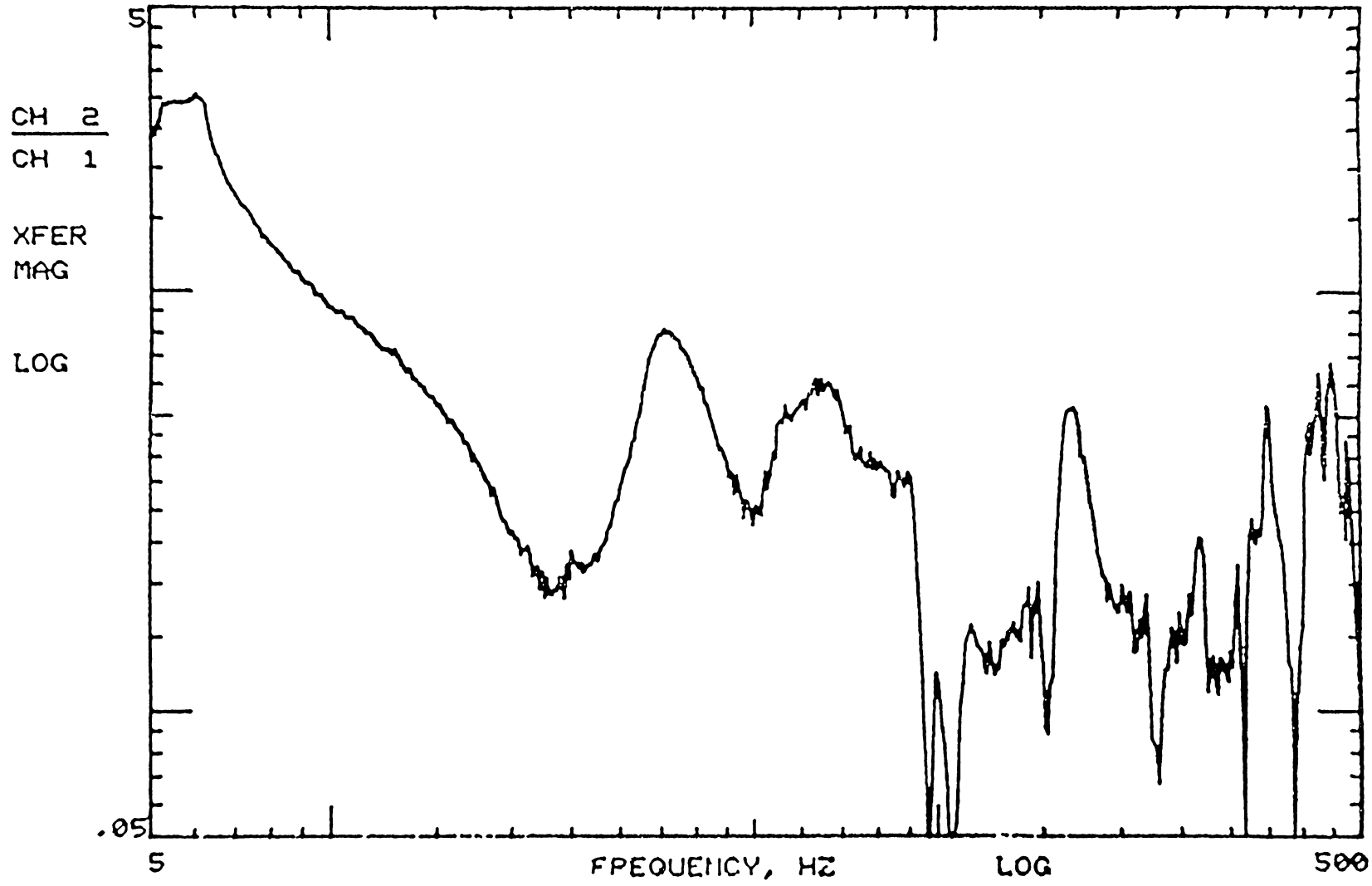


CODAR TECHNOLOGY MILITARY MICROVAX

.5G SINE X-AXIS 18-DEC-86 MODEL 600M SPERRY #8

6 shocks AM

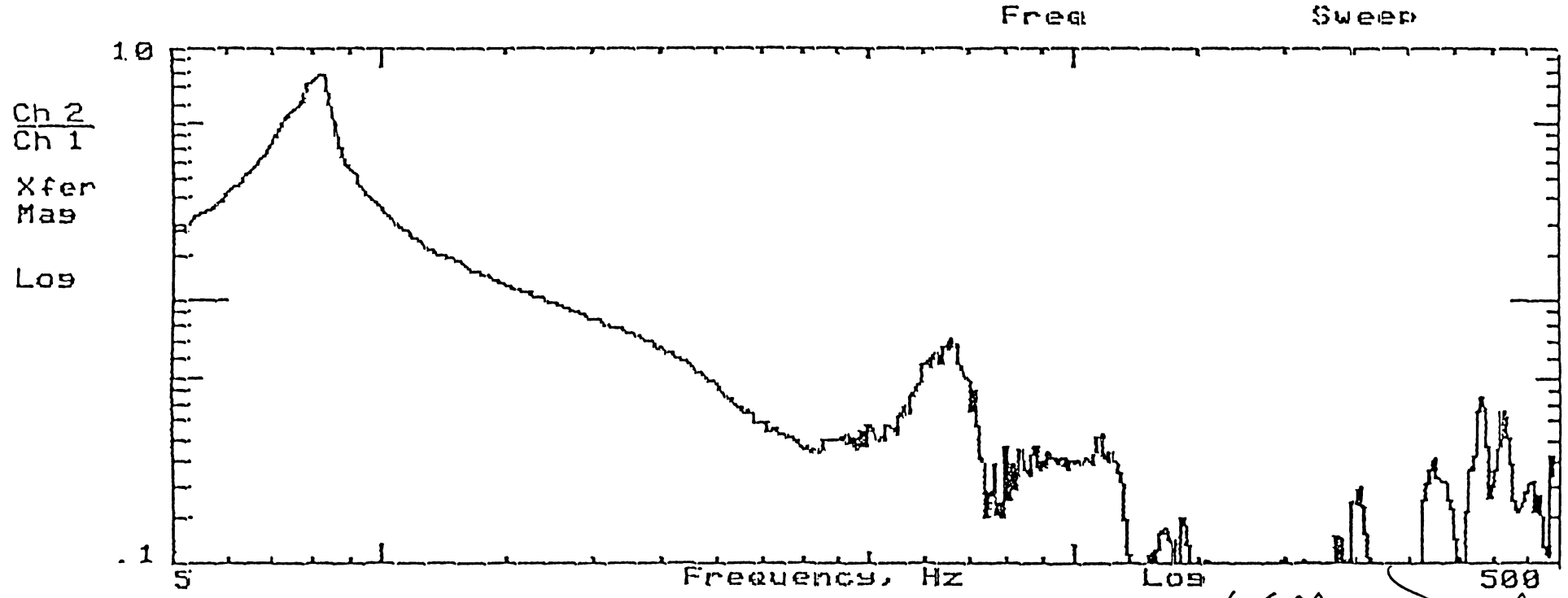
CODAR Technology, Inc.
600M Shock and Vibration Data
Ball Aerospace Corporation 12/86
DATA RUN Number: 1
*** Copyright (C) 1986 ***



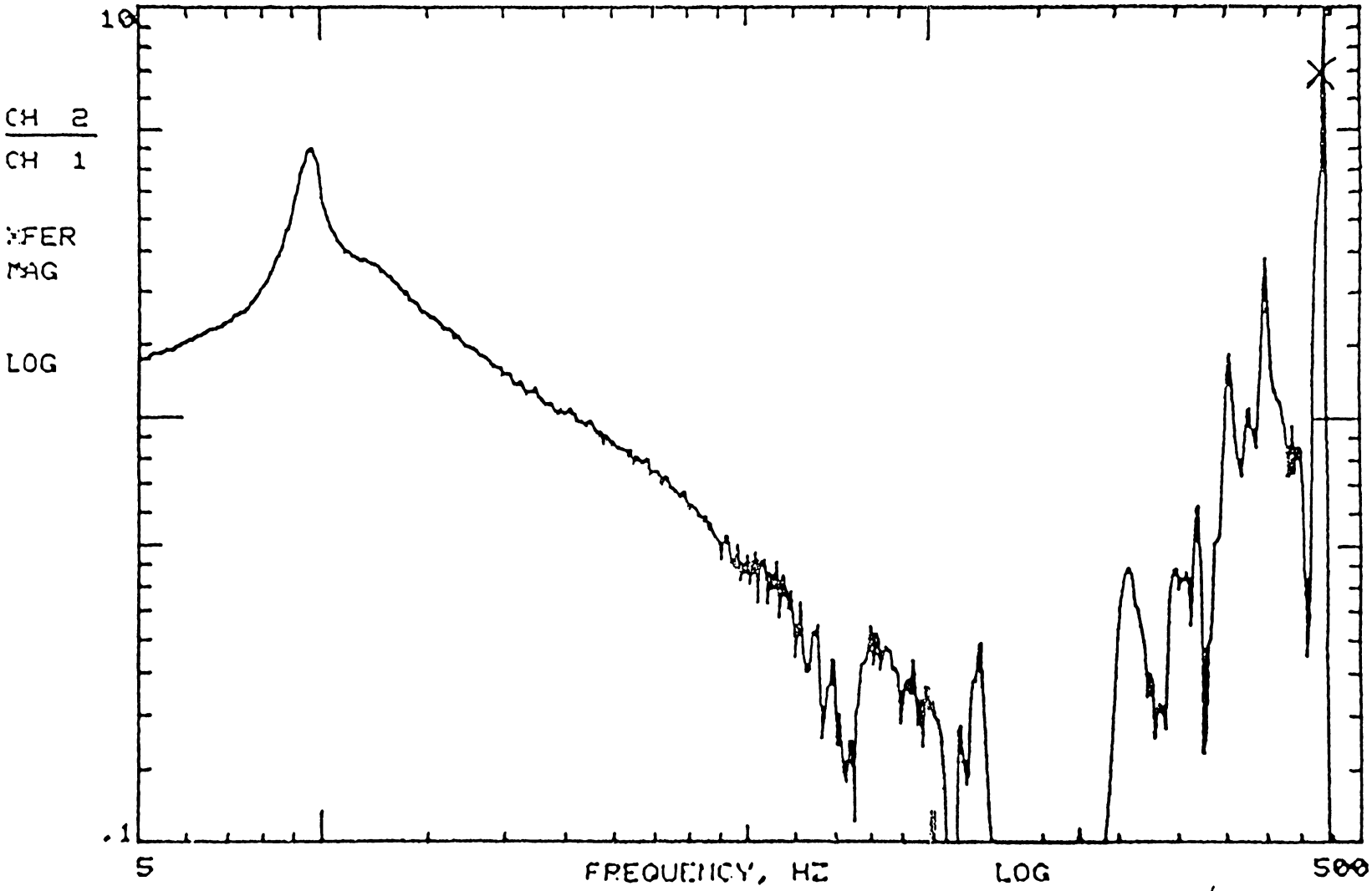
CODAR TECHNOLOGY MILITARY MICROVAX DATA/X DISK DRIVE
.50 SINE X-AXIS 18-DEC-86 MODEL 600M SPERRY #2

6 Shock, 11ms

CODAR Technology, Inc.
600M Shock and Vibration Data
Ball Aerospace Corporation 12/86
DATA RUN Number: 2
*** Copyright (C) 1986 ***



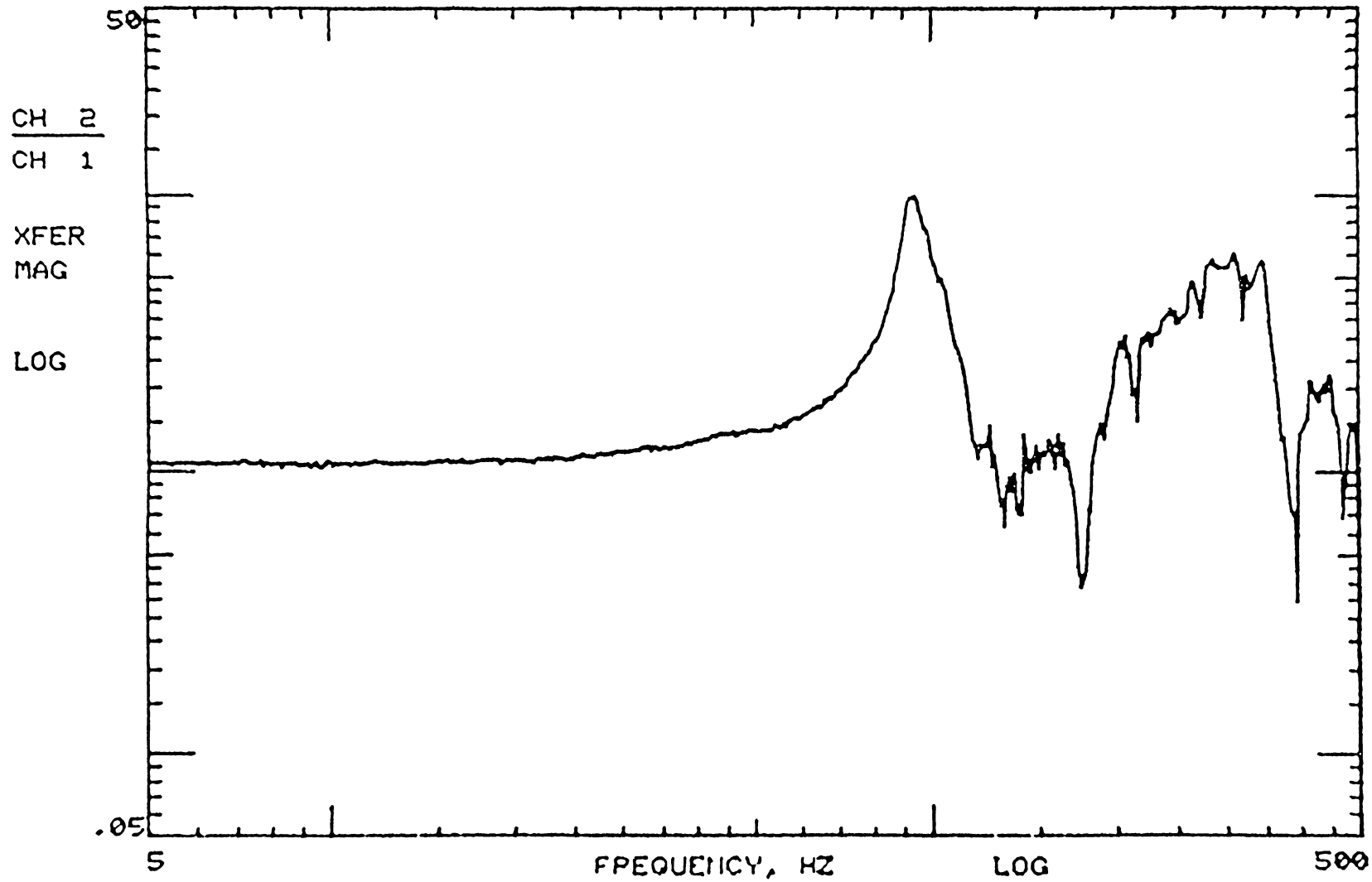
CODAR Technology, Inc.
600M Shock and Vibration Data
Ball Aerospace Corporation 12/86
DATA RUN Number: 32
*** Copyright (C) 1986 ***



CODAP TECHNOLOGY MILITARY MICROVAX 600M
.5G SINE SWEEP Z-AXIS 22-DEC-86 SPERRY # 2 65M

DATA IZ DISC TRAY

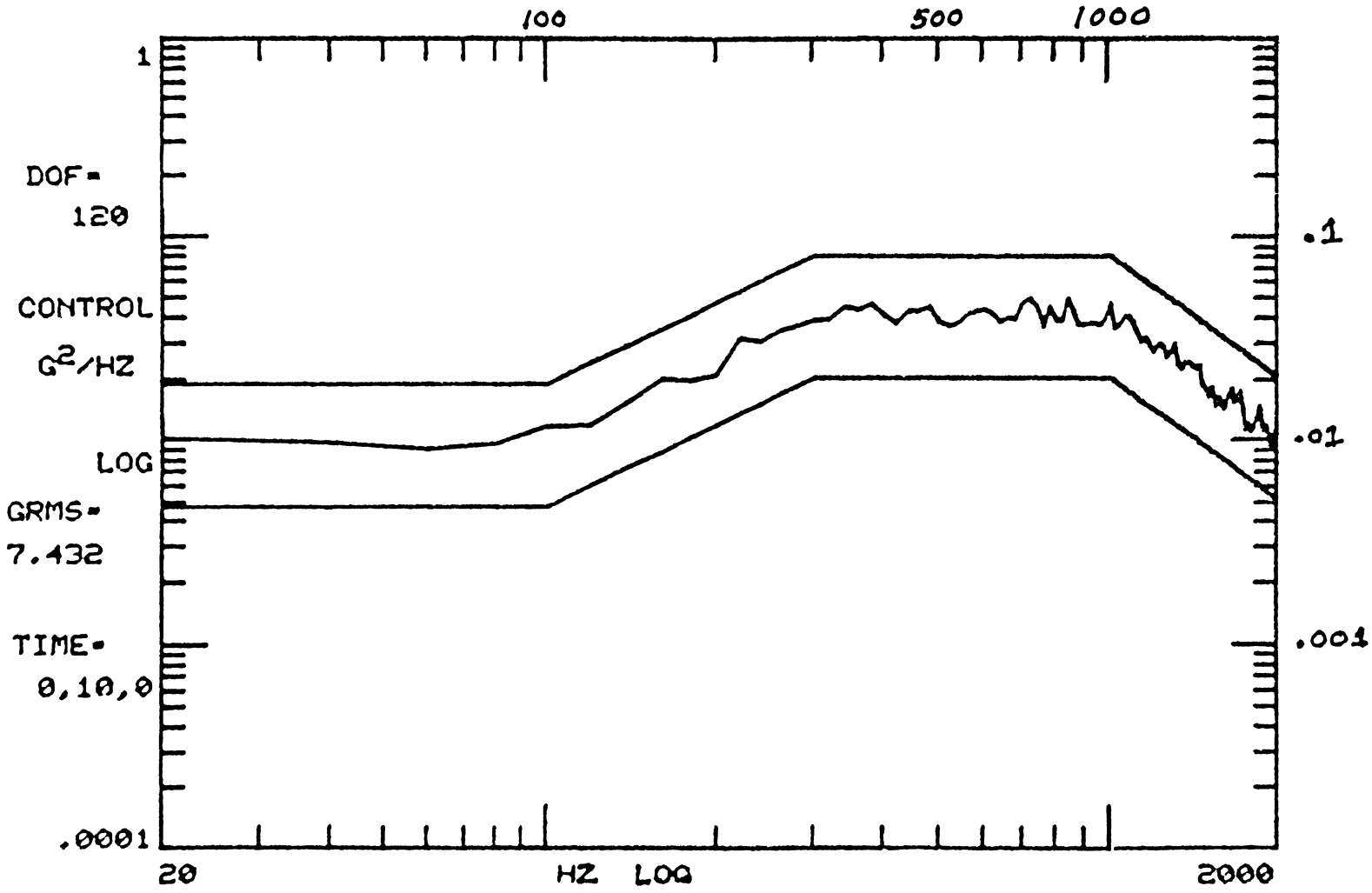
CODAR Technology, Inc.
600M Shock and Vibration Data
Ball Aerospace Corporation 12/86
DATA RUN Number: 49
*** Copyright (C) 1986 ***



CODAR TECHNOLOGY MILITARY MICROVAX DATA 2X BACK PLANE
.5G SINE X -AXIS 18-DEC-86 MODEL 600M SPERRY #2 BOARD

6 shots AM

CODAR Technology, Inc.
600M Shock and Vibration Data
Ball Aerospace Corporation 12/86
DATA RUN Number: 3
*** Copyright (C) 1986 ***

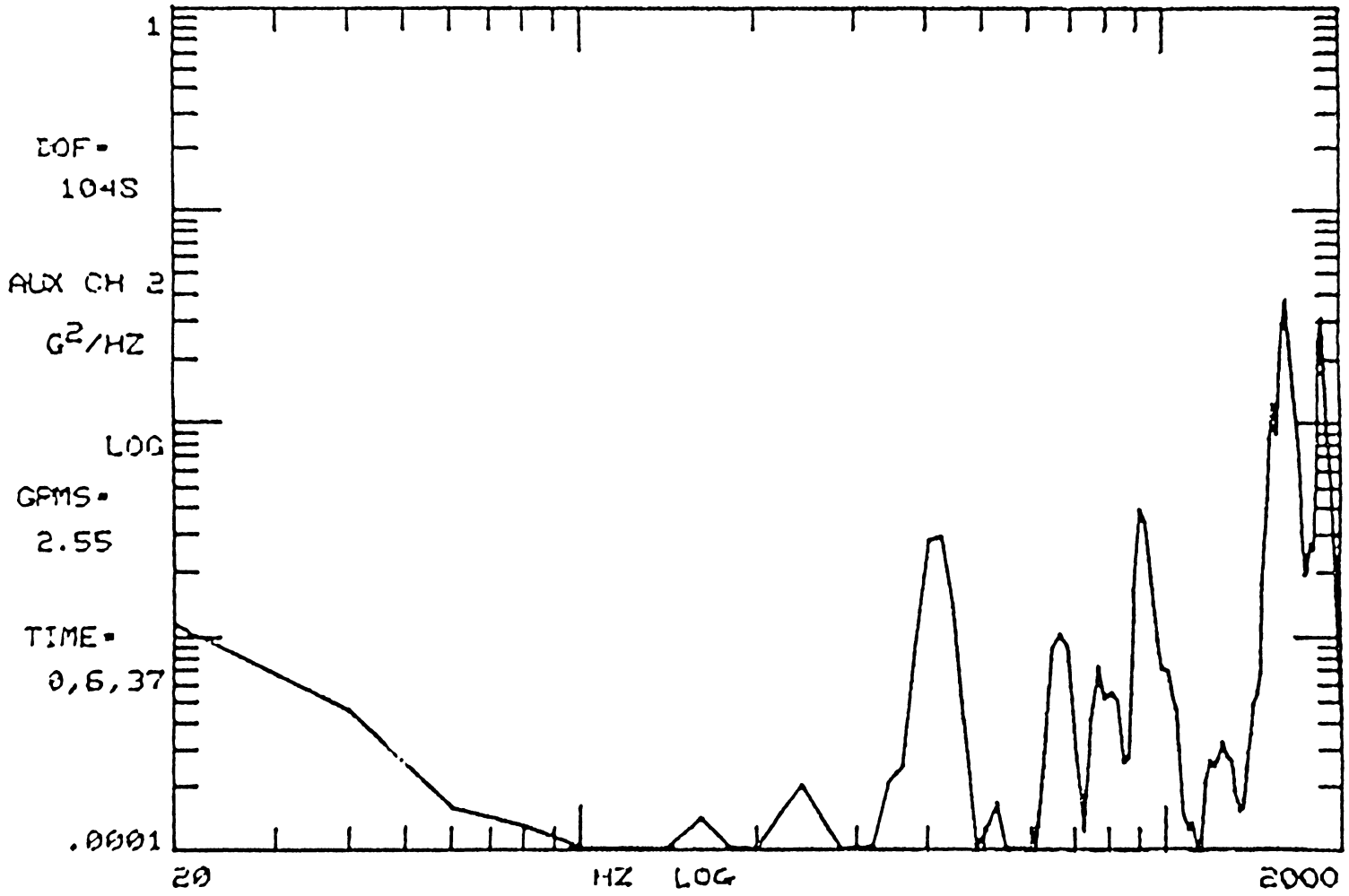


CODAR TECHNOLOGY MILITARY MICROVAX 8 MOUNTS
RANDOM X-AXIS MODEL 600M SPERRY # 2

30 G PEAK
810C JET AIRCRAFT RANDOM V.
7.5

12/13/86
PM

CODAR Technology, Inc.
600M Shock and Vibration Data
Ball Aerospace Corporation 12/86
DATA RUN Number: 25
*** Copyright (C) 1986 ***

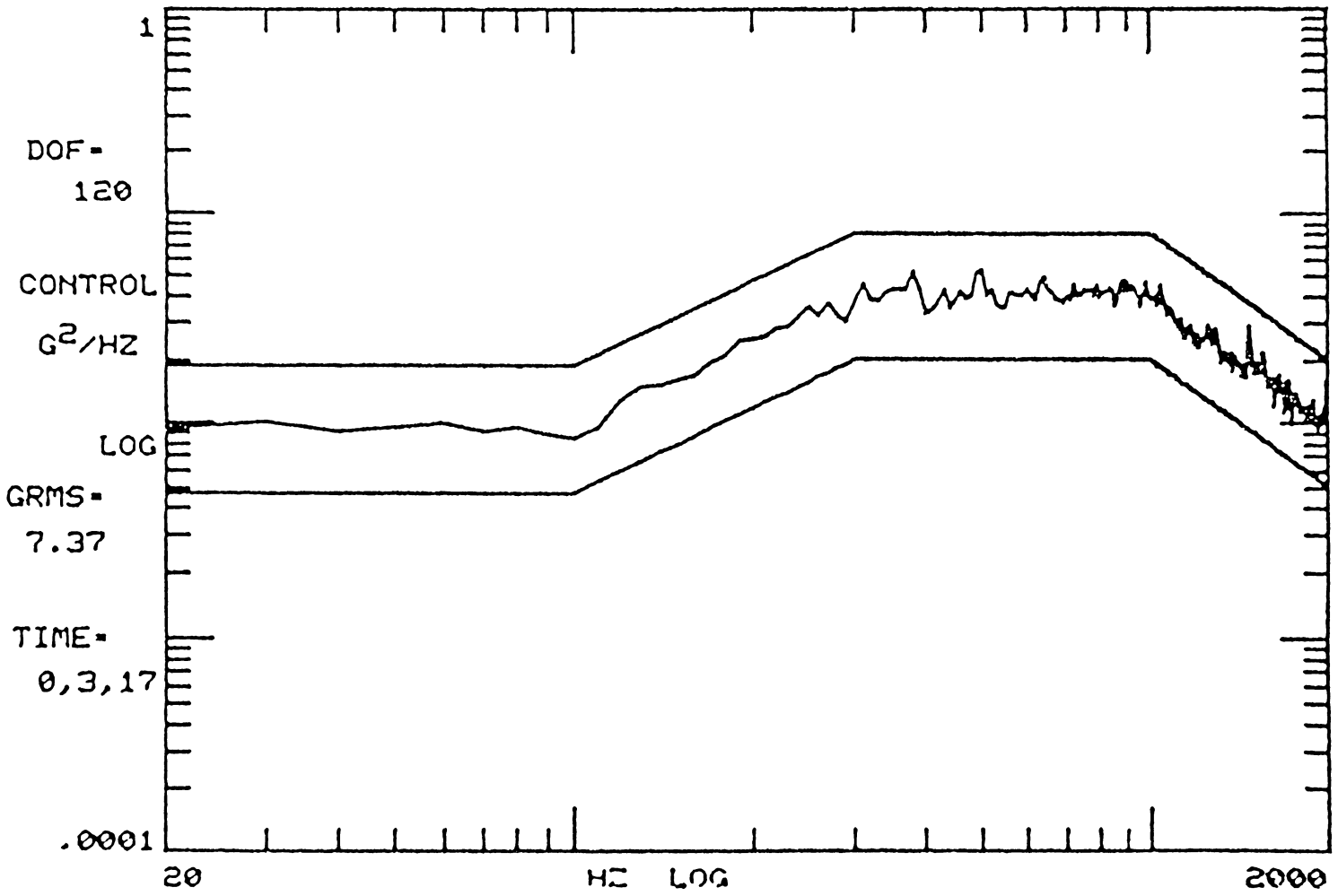


CODAR TECHNOLOGY MILITARY MICROVAX 8 MOUNTS
RANDOM X-AXIS MODEL 600M SPERRY # 2 Data 1

CODAR Technology, Inc.
600M Shock and Vibration Data
Ball Aerospace Corporation 12/86
DATA RUN Number: 26
*** Copyright (C) 1986 ***

12/86 12/18/86
PM

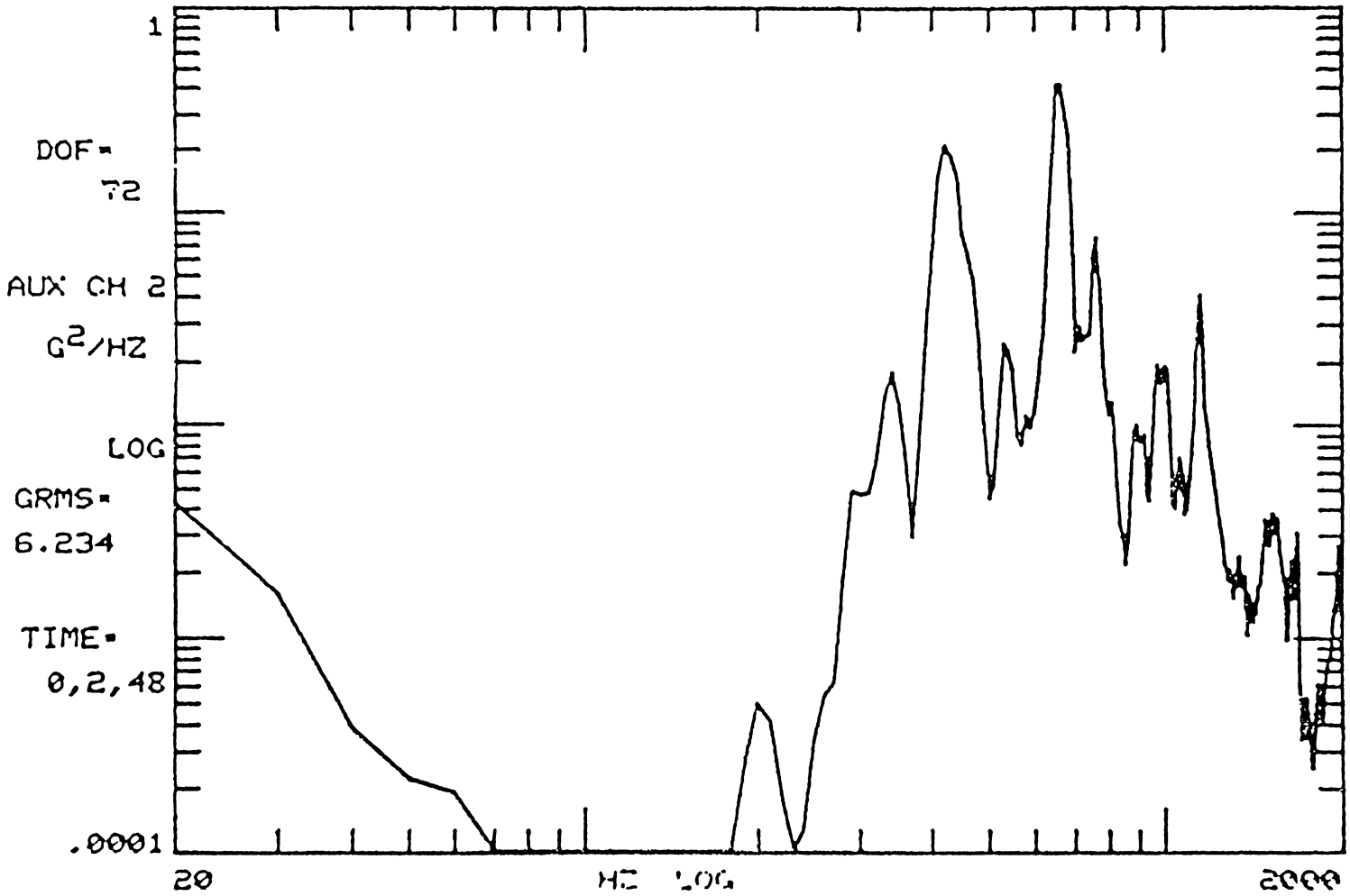
(-U



CODAR TECHNOLOGY MILITARY MICROVAX

RANDOM Z-AXIS 600M SPERRY #2 22-DEC-86

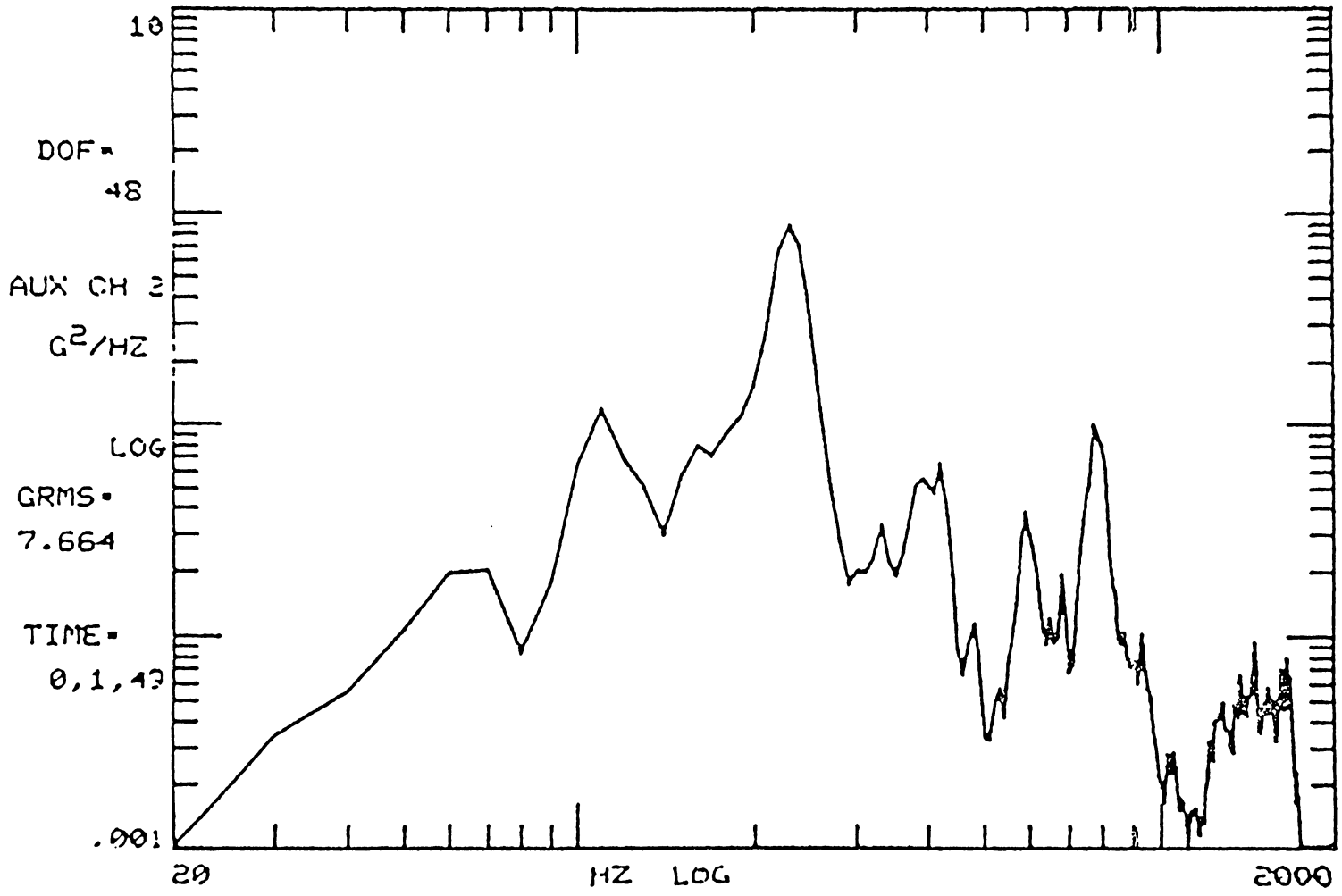
CODAR Technology, Inc.
600M Shock and Vibration Data
Ball Aerospace Corporation 12/86
DATA RUN Number: 66
*** Copyright (C) 1986 ***



CODAR TECHNOLOGY MILITARY MICROVAX DATA 12 DEC 1986

RANDOM Z-AXIS 600M SPERRY #2 22-DEC-86

CODAR Technology, Inc.
600M Shock and Vibration Data
Ball Aerospace Corporation 12/86
DATA RUN Number: 67
*** Copyright (C) 1986 ***

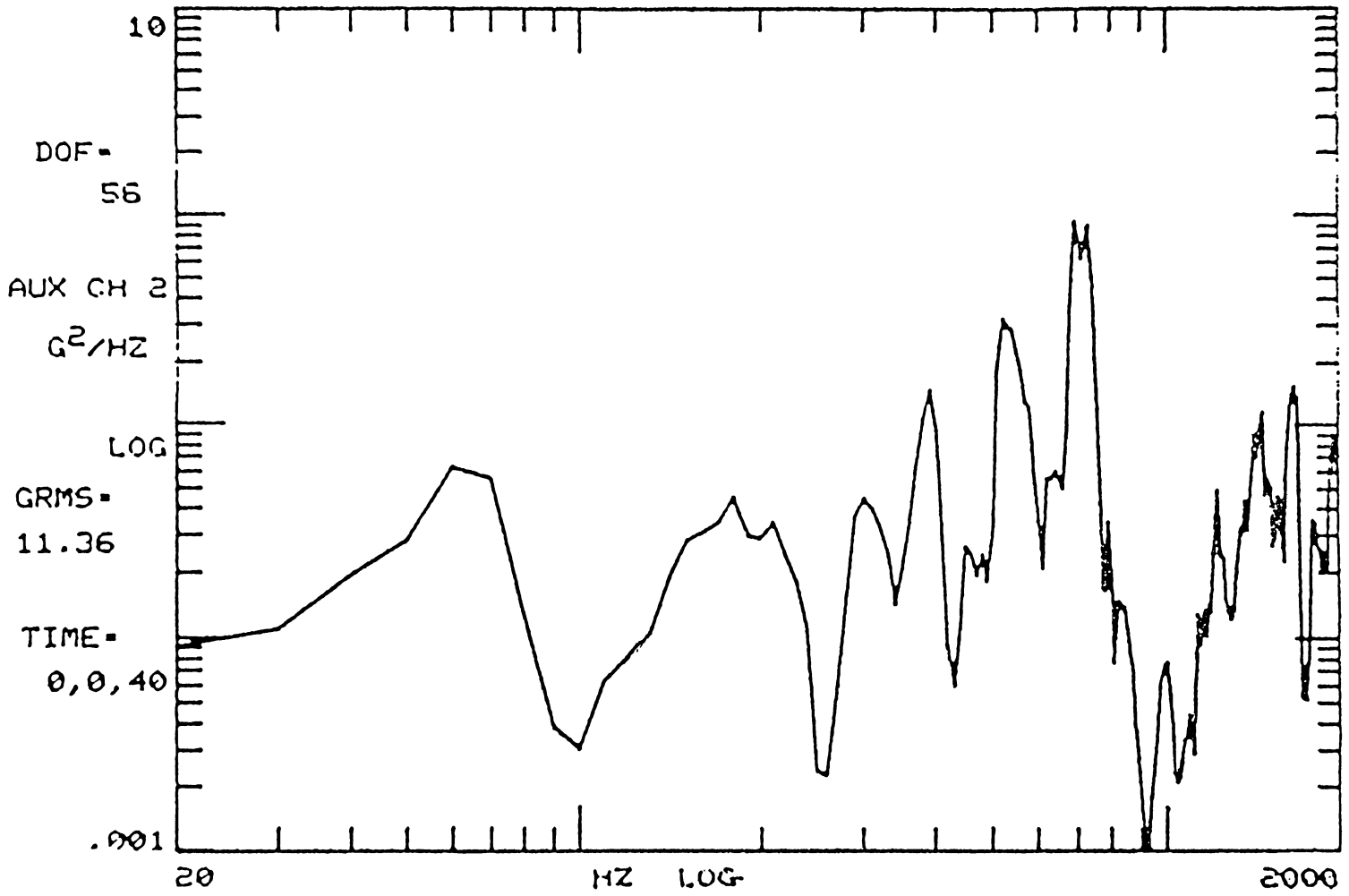


CODAR TECHNOLOGY MILITARY MICROVAX
RANDOM Z-AXIS 600M SPERRY #2 22-DEC-86

Data by Board

CROSS TALK

CODAR Technology, Inc.
600M Shock and Vibration Data
Ball Aerospace Corporation 12/86
DATA RUN Number: 68
*** Copyright (C) 1986 ***



CODAR TECHNOLOGY MILITARY MICROVAX
RANDOM Z-AXIS 600M SPERRY #2 22-DEC-86

DATA BOX POWER SUPPLY

CODAR Technology, Inc.
600M Shock and Vibration Data
Ball Aerospace Corporation 12/86
DATA RUN Number: 69
*** Copyright (C) 1986 ***

BLANK

APPENDIX C:

TECHNICAL REFERENCE PAPER

BLANK

Severe Service Requires Strenuous Shock/Vibration Testing

by Chet Gilbert and Herb LeKuch

Reproduced with permission from November, 1982
MILITARY ELECTRONICS/COUNTERMEASURES Magazine.

Severe Service Requires Strenuous Shock/Vibration Testing

by Chet Gilbert and Herb LeKuch

Shock and vibration rank among the most destructive agents in our industrial society. The challenge that the design engineer faces is to recognize the complexities of providing effective isolation, particularly in severe service where the stakes are high and where protection against equipment failure due to inadequate shock and vibration isolation must be absolute.

Today severe service conditions are being encountered individually and in combination in a wider spectrum of applications such as shipboard electronics, avionics, chemical processing, geophysical and ordnance equipment, over-the-road vehicles, and shipping containers. The parameters of severe service applications are identified in Figure 1.

Designers working in these industries are on the alert for isolation systems that will provide the most practical defense against the damaging effects of shock and vibration. Resiliently mounted equipment must be protected from these forces in its environment, and from harming its own surroundings. Ideally, a passive isolation system requires no maintenance, takes little space, and keeps functioning under varying temperatures and in corrosive environments. It also should have the strength to restrain its load under catastrophic conditions and to return it to its normal operating position after the deflections caused by repeated shock loadings.

Each day, applications engineers field numerous questions involving isolation of shock and vibration in severe service. We have compiled a list of the 20 most often asked questions and have attempted to supply unbiased answers.

Many of the questions come from engineers who have little experience in isolation technology or who have been away from the subject for many years. Their questions are on "the basics."

1. Why is shock and vibration isolation such an important subject today?

Shock and vibration influence the operation, performance, and life expectancy of almost every type of electronic (as well as mechanical) equipment. If not correctly predicted and compensated for, they may erode the life of the equipment, driving it from service long before its time. Sometimes this premature deterioration stems from just a few impact loadings, sometimes from many fatigue cycles of vital parts, and sometimes from a combination of both.

2. What is considered severe service?

Severe service environments are those where destructive shock and vibration forces are experienced and also where extreme temperatures,

Chet Gilbert is vice president, Isolator Products; Herb LeKuch is program manager; both authors are affiliated with Aeroflex Laboratories, Inc., 35 South Service Road, Plainview, NY 11803; (516) 694-6700.

Application	Types of Equipment Protected	Sources of Vibration and Shock	Other Hazards of the Environment	Critical Specifications (Limitations)	Needed Isolator Characteristics	Other Requirements
Shipboard Electronics	Navigation Displays, Fire Control, Switch Gear	Nearby Blast, Ship Vibration	Salt Water, Temperature Extremes	MIL-S-901C MIL-STD-167	Life of Ship Equipment	Can Be Painted; Compliant in All Directions
Over-the-Road Vehicles	Instrumentation Housings, Generators	Irregular Terrain	Temperature Extremes, Ozone, Radiation	Munson Rough Road Course; 10 g's Repeated Shock	Long Fatigue Life, Large Displacement	Minimum Space, Sometimes Inaccessible After Installation
Shipping Containers	Jet Engines, Missiles	Transit, Handling Drop	Altitude Variations, Exposure to Moisture	Accidental Drop from 48 Inches; ATA 300	Excellent Shock Mitigation	Indefinite Shelf Life; Repeated Use
Geophysical Equipment	Data Acquisition, Data Processing Electronics	Over-the-Road Vehicles, Transit, Ship Loading	Misaligned Installation	Severe Road Shock, Careless Handling, Sustained Road Vibration	Maintenance-Free, No Replacement	Repeated Large Deflections Due to Load Shock
Chemical Processing Equipment	Centrifuge, Dryers, Pumps	Unbalanced Dynamic Loads	Corrosive Environments, Chlorine, Sulfur	High Temperature and Corrosive Environments	Low Frequency Response	Maintenance Free
Avionics	ECM, Communications, Reconnaissance	Rapid Maneuver, Hard Landings	Temperature and Altitude Extremes	15 g, 11 ms Hard Landing MIL-STD-810	Long Fatigue Life, No Aging Deterioration	Low Profile; Dynamic Response Doesn't Change with Temp. or Altitude
Ordnance Equipment	Missile Launcher, Tank Artillery, Computer Control	Off-Road Vehicles, Railroad Humping	Nearby Blast, Temperature and Climate Extremes	Munson Rough Road; Railroad Humping, 7 Fps	Excellent Shock Mitigation	Can Be Painted; Used in Any Altitude; Long-Term Storage

Figure 1. Parameters of Severe Service Applications

ranging from as low as -400° F to as high as +700° F, are encountered. The environment may include contaminating and destructive elements, such as ozone, oil, grease, sand, salt spray, and organic solvents that insidiously attack isolator materials.

Other physical parameters of a specific application can contribute to the severity of the service environment. Severity increases:

- Where unusually heavy static weight is to be isolated. A single isolator may be asked to handle 5,000 pounds or more.
- In compact, low-profile envelopes, where space is at a premium. The challenge becomes even more severe when required to replace a failed isolator with an alternate.
- Where equipment needs effective 3-plane, all-axis—vertical, horizontal, lateral—isolation.
- Where isolators must be position-insensitive, working in compression, tension, shear, or roll.
- Where a single isolator must handle both shock and vibration.

3. What is shock? vibration?

An object receives an impulse or shock when it experiences a suddenly applied force. The impulse is relatively short; its duration is generally in milliseconds, and its magnitude may be measured as the relative change in velocity or acceleration of the object. An isolation system will minimize the induced loads by first absorbing the energy of the high acceleration, short duration impulse, and then dissipating it at a lower magnitude but over longer duration, thereby avoiding damage.

Vibration is a different type of disturbance. *Sinusoidal* vibration is a harmonic disturbance or oscillating motion described by the amplitude and frequency of the variable force. Isolator efficiency can readily be determined by comparing amplitude (input force or displacement versus output) at any frequency. At the higher frequencies, the response will be substantially less than the magnitude of the applied force.

Random vibration is a condition where the disturbance is highly irregular and, because of its complication, can be thought of as being composed of a broad spectrum of constantly varying frequencies and amplitudes. The effectiveness of isolation is measured by the reduction in the mean square value of vibration. The isolation system filters out much of the higher frequencies and broadens the response in the region of its natural frequency.

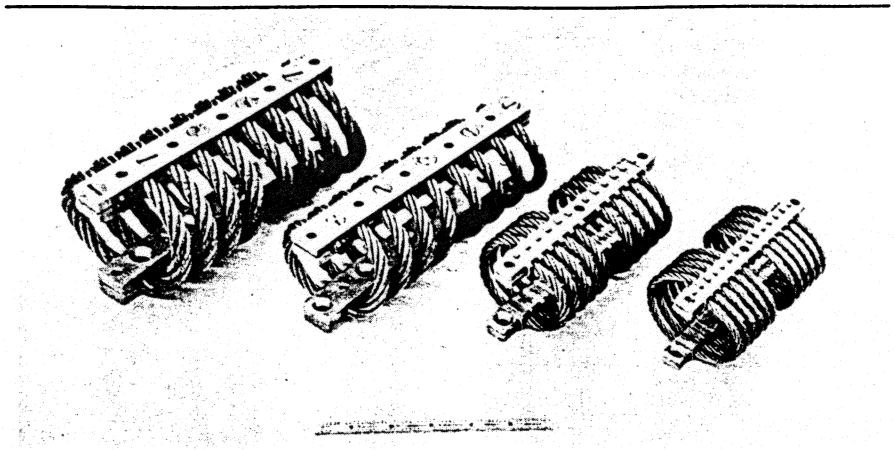


Figure 2. Helical Cable. Each helical cable isolating element has specific response characteristics determined by the diameter of the wire rope, the number of strands, the cable length, twist or lay, and number of cables per section. Inherent damping is provided by flexure hysteresis (the rubbing and sliding friction between the strands of rope).

Isolators are used as an interface to protect sensitive equipment from the dynamic forces and motion of vibration and shock environments. They function to attenuate the impressed forces by virtue of their controlled elasticity and damping characteristics.

4. What types of isolator and isolator systems are available?

A typical vibration or shock isolator consists of a resilient element generally housed in a metallic supporting frame. Elastomers, air, steel springs, wire rope, and metal mesh are the most common types of resilient elements in use today. Of all of these, the serious contenders for severe service applications are elastomers and wire rope.

- *Elastomeric Isolators:* Molded of natural or synthetic rubbers, their design primarily involves stressing the elastomer in either shear or compression. Elastomers are molded to the size or shape and stiffness to fit individual applications. The resiliency of elastomers makes them suitable as a material for vibration isolators, but they have limitations in temperature and mechanical stress. The approximate stable temperature extremes for most elastomer isolators is in the range of -65° F to +165° F. A wider temperature range may result in surface cracks under stress as well as changes in characteristics.

Under continuous stress reversals involving a high deflection magnitude, elastomers perform initially well, but tend to drift in their stress/strain characteristics over a period of time. It is advisable to limit strain to approximately 20 percent in compression and under 50 percent in shear to expect reliable operation.

- *Wire Rope Isolators* (see Figure 2): Helical wound cable isolators are assemblies of stranded steel wire rope formed between metal retainers. Their large dynamic displacement attenuates heavy shocks, while their inherent damping enables them to absorb and dissipate large amounts of low and high frequency vibration. Of particular importance for motion control at resonance is their flexure hysteresis.

Damping characteristics, regardless of the direction of applied force, are related to the strain applied to the isolator. Both large and small amplitude motions are moderately damped with the isolator appearing to act as a nonlinear element. They provide effective protection in compression, tension, shear, or roll.

Helical cable isolators are made entirely of stainless steel and corrosion-resistant aluminum alloy; or even more exotic metals such as Inconel. They function from -400° F to +700° F and resist ozone, oil, grease, sand, salt spray, and organic solvents.

5. What is the effect of temperature, drift, creep, etc., on the dynamic performance of isolators?

Because equipment has to be protected over a wide temperature range, the ability to calculate the isolation system effectiveness at the

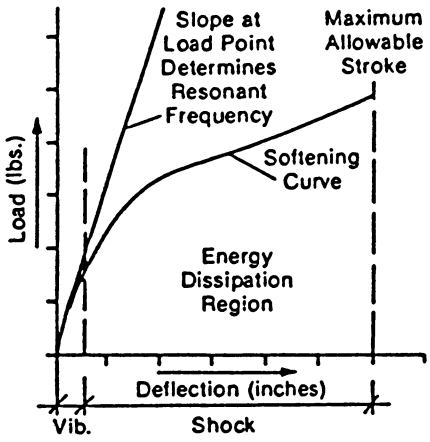


Figure 3. Load Deflection Curve. This curve of nonlinear isolator in compression shows performance in the ability to deflect while isolating the resonant frequency.

low and high temperatures is important. With elastomers, temperatures nearing -20° F and below will harden the material, shifting the response to a higher frequency as the stiffness increases. Temperatures above approximately 140° F will shift the frequency down with greater amplitude at resonance due to material softening. An increase in stiffness or amplitude can lead to early damage of the object due to more vibration experienced at critical components.

Helical isolators using CRES wire rope are essentially insensitive to temperature extremes with the response at room conditions typical of what would be experienced after conditioning.

Drift and creep also affect performance. An elastomer will experience a small percentage in thickness loss under stress. There may be thickness loss due to repetitive shock or vibration stresses, and long-term thickness loss due to static stress. Because there is a physical change in thickness, the dynamic stiffness and shock response as well as the damping properties may be degraded. In selecting an isolator, evaluate what the long-term drift and creep characteristics are. To assure performance, the use of an isolator whose properties remain relatively constant is preferable. Tests with stabilized wire rope isolators result in negligible creep under load, and minimal set due to repetitive load. With CRES wire, particularly at temperature extremes, there is no significant change in performance.

6. What are the two most essential properties of effective isolators for severe service?

Shock and vibration so often occur together that an isolator must have two seemingly incompatible properties to counteract both forces effectively. These are a low natural frequency, to isolate the relatively high frequency vibrations that lead to fatigue-caused structural damage; and the ability to deflect in a controlled and repeatable way to absorb the impact loadings that would otherwise be transmitted into the isolated unit.

A nonlinear isolator with a softening load profile and amplitude-dependent damping meets both of these requirements. A system with helical wire rope isolating elements, properly designed, exhibits far less than half the critical frequencies of the sensitive equipment. Elastic yielding of the helix under heavy shock loadings without bottoming or permanent set, provides the large deflections needed to bring the impacts within acceptable limits. The load/deflection graph (see Figure 3) shows these two distinct regions of isolator performance.

7. Is there any one isolation system that handles both shock and vibration?

The nonlinear wire rope isolator handles both shock and vibration. Referring to Figure 3, in the vibration region, the isolator is relatively stiff. As the amplitude of vibration increases, the mount becomes progressively softer, shifting the resonant frequency away from the input "driving" frequency and isolating the system.

The nonlinear cable mount softens as a shock load is applied. As it deforms, it absorbs energy. At approximately 75 percent of its compression stroke, it begins to stiffen. However, the displacement has already reduced the acceleration level, dissipating the energy through the controlled buckling of the helical loops.

A hardening isolator would stiffen rapidly as the shock load is applied. The helical shows more nearly uniform energy dissipation over its entire deformation, keeping the maximum deceleration within acceptable limits. It is almost a constant force absorber over most of its stroke, regardless of how rapidly the load is applied. This characteristic means that its performance is predictable and easily confirmed.

8. What are the most important and desirable requirements when considering isolators for application in severe service?

A few of the desirable aspects of severe service isolators are:

- *A practically unlimited load range:* Heavy isolating mounts can be attached to almost any machine or enclosure, no matter what its weight. Static load ratings that exceed 15,000 pounds are often needed.
- *A compact, low-profile envelope:* The need to attenuate heavy shock loadings with minimum deflection often requires isolators where space is at a premium, an advantage in new equipment and sometimes a necessity when replacing other isolating media that have failed in service.
- *A three-plane, all-axis isolation:* Equipment may have to be free to move in any direction on isolators—vertically, horizontally; and laterally.
- *Ability to work in any position:* Isolators may require loading in compression, tension, shear, or roll in base mount, CG mount, or cantilevered mount. They should be available for application anywhere in an isolating system.

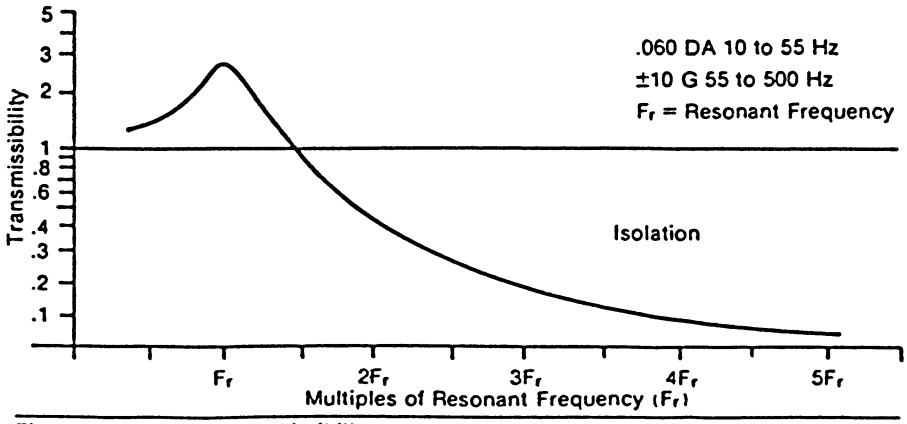


Figure 4. Vibration Transmissibility

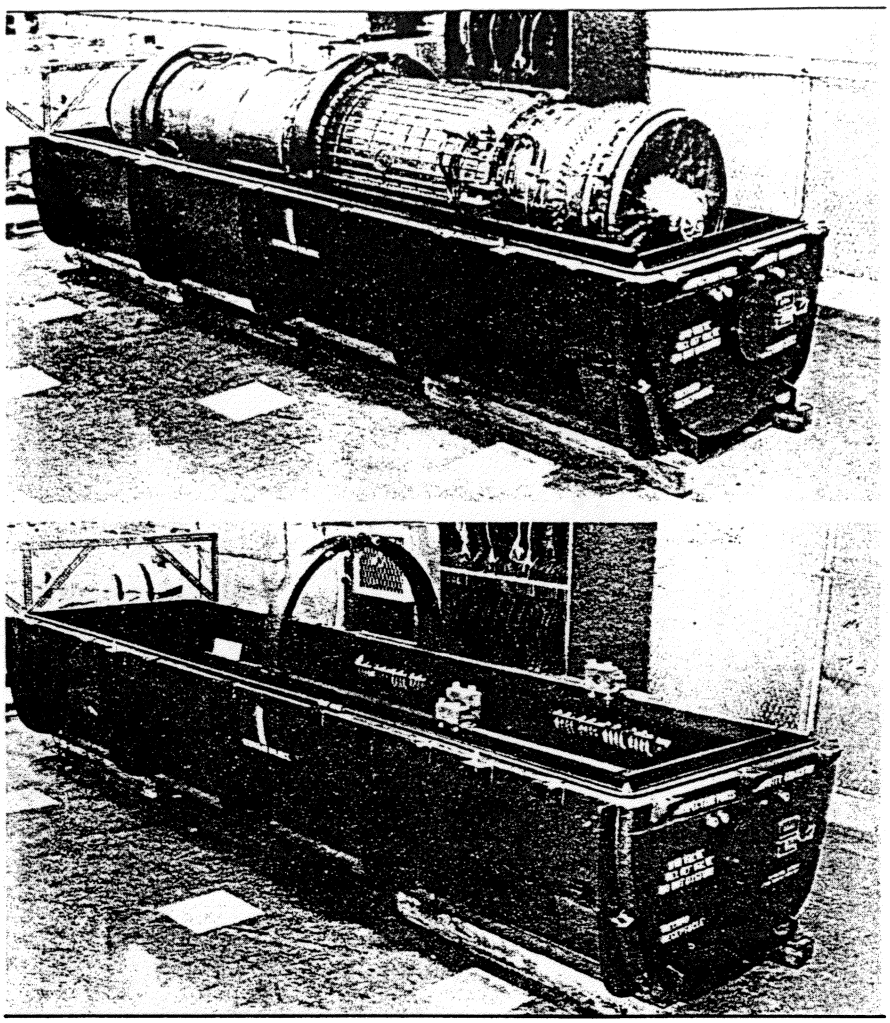


Figure 5. A Jet Engine Shipping Container. The container carries its load frame isolated on helical cable mounts to save space. Metal construction promises virtually unlimited service life and also provides electrical conductivity; the need for ground straps is therefore eliminated.

• *Resistance to destructive environments:* They should be impervious to environmental attack, and require little or no maintenance.

9. How do you measure or predict the level of protection needed?

If you want to predetermine isolation requirements early in the design program, obtain from your supplier a computer analysis of shock and vibration to predict the type of mount or mounting system that will be needed to meet the predicted parameters. Using such inputs as package geometry, weight, center of gravity, fragility level, and available sway space, you can calulate the spring rate, dynamic travel, and natural frequency required to provide protection under specific ranges of dynamic conditions. A typical vibration transmissibility curve is shown (see Figure 4).

10. Are there any special considerations when shock is the dominant problem?

Isolators must have the ability to attenuate shock with minimum dynamic travel—a level of shock protection often unavailable from other than the cable isolating medium. They make possible very compact mounting systems in shipping containers carrying sensitive loads, for vehicle-carried electronic instruments, and for many other installations where the equipment must be protected from dynamic forces inherent in its environment (see Figure 5).

Typically, shock is much more severe than vibration and isolators must have enough deflection capacity to absorb shock without bottoming.

The nonlinear response of a helical isolator dissipates far more energy in the same travel or stroke than a linear device. In worst-case situations, 75 percent of the allowable stroke satisfactorily alleviates the shock with an adequate margin of safety.

11. Are there any special considerations when vibration is the dominant problem?

Where vibration is the dominant problem, such as in rotating or reciprocating machinery, understanding the forces involved and knowing how they are measured is most important:

• The forcing frequency (F_f) is the major vibrational frequency of a machine, expressed in cycles per second, or hertz (Hz).

• The frequency ratio is derived by dividing the machine's forcing frequency by the isolating system's natural frequency. The isolating efficiency of the system depends on it, and is expressed as a percentage of the vibration force at the forcing frequency that is blocked by the isolators.

• The natural frequency of a system is the rate at which it oscillates as it returns freely to rest after having been displaced or agitated by an outside force. It can be calculated by the equation:

$$F_N = 3.13\left(\frac{1}{S_{ST}}\right)^{1/2} \text{ or } F_N = 3.13\left(\frac{K}{W}\right)^{1/2}$$

where S_{ST} = Static deflection
K = Stiffness
W = Weight

• Resonance occurs when the natural frequency of the mounting system coincides with the forcing frequency. If it lasts for any length of time, it can be very dangerous because the amplitudes of the vibrations will become excessive.

• Damping is the capacity of the isolating element to absorb part of the mechanical energy applied by the forcing frequency and convert it to heat.

12. Does isolation have to be 100 percent?

Effective vibration isolation needn't be 100 percent. Often, total isolation may require large static deflections. This means large isolators with increased space requirements and higher costs. Isolation of about 80 percent is usually satisfactory.

Fatigue can be avoided by reducing the amplitude of response stress cycles. Here, the isolator protects by reducing the actual level of vibration

in the upper frequencies where the cumulative number of cycles would otherwise approach failure criteria.

The performance of an isolator in vibration control depends on the applied load and direction of force. Most helical isolation systems provide an amplification of 3 to 3.5 at resonance. Most other commercial isolators exhibit amplification of 7 or 8 at resonance.

13. What information is needed to select the isolation system when shock dominates?

For specific shock applications:

- Determine the static load each mount will carry.
- Estimate the fragility of the load in terms of the number of g's you believe it can absorb without damage. Many years of designing isolating systems permit us to establish a maximum tolerable impact range of 15 to 25 g's for survival of a fragile load. Military equipment will usually withstand a 25 to 35 g single-shock event, but commercial equipment should be protected to the lower end of the range.
- Using the same load-carrying criteria that you would for vibration isolation, refer to the load/deflection curves. Verify that the calculated deflection (based on the anticipated shock response in g's, to static load and a scaling factor) is less than the maximum allowable deflection of the isolator.
- More than one isolator may have suitable load/deflection curves so that your choice can be dictated by secondary considerations such as size and price. A general rule in the absence of any overriding requirement is to select the mount whose expected deflection due to shock is approximately 70 to 75 percent of its maximum allowable deflection.

14. What information is needed to select the isolation system when vibration dominates?

The following steps may be followed when selecting helical cable isolation mounts if vibration is the dominant problem. Specific information on isolator types and load/deflection curves will be required. This material is available from the manufacturer. If selecting elastomer mounts, similar steps are followed, and the manufacturer's literature may provide specific guidelines.

- Determine the static load each isolator in the mounting system will have to carry; i.e. divide total weight by the number of mounting points.
- Determine the machine's lowest forcing frequency.
- Determine how much isolation you need. Except for special cases, 80 percent isolation will usually be enough.
- Find the static deflection for the forcing frequency and the percent of isolation. Use the previously mentioned Frequency/Static Deflection equation.
- Select the appropriate isolators for the static deflection and maximum loading. If shock loading may occur, select from those isolators whose static load rating is at least five times as high as the applied static load.
- Now refer to the load/deflection

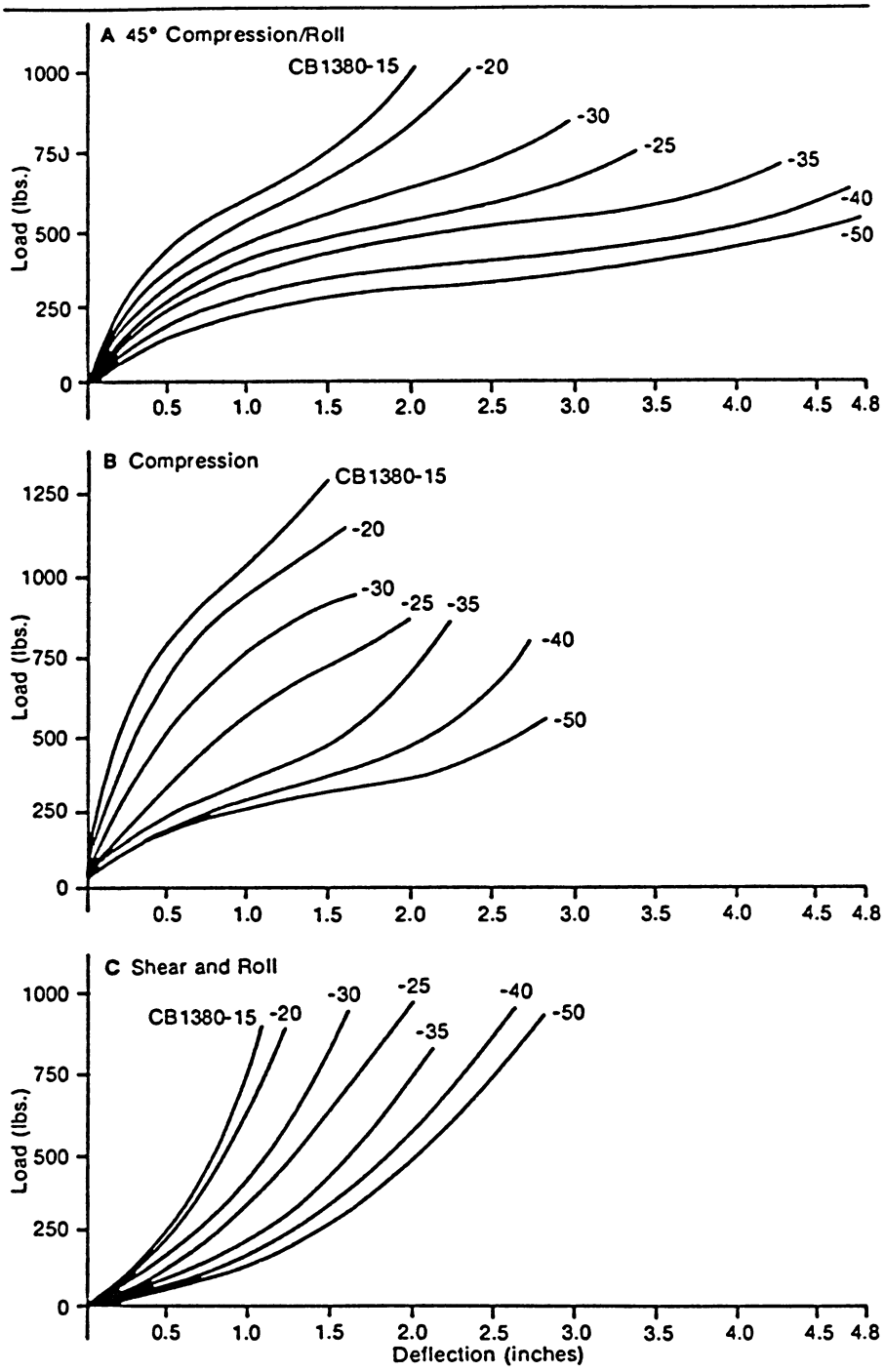


Figure 6. (a) 45° Compression Roll (b) Compression (c) Shear and Roll

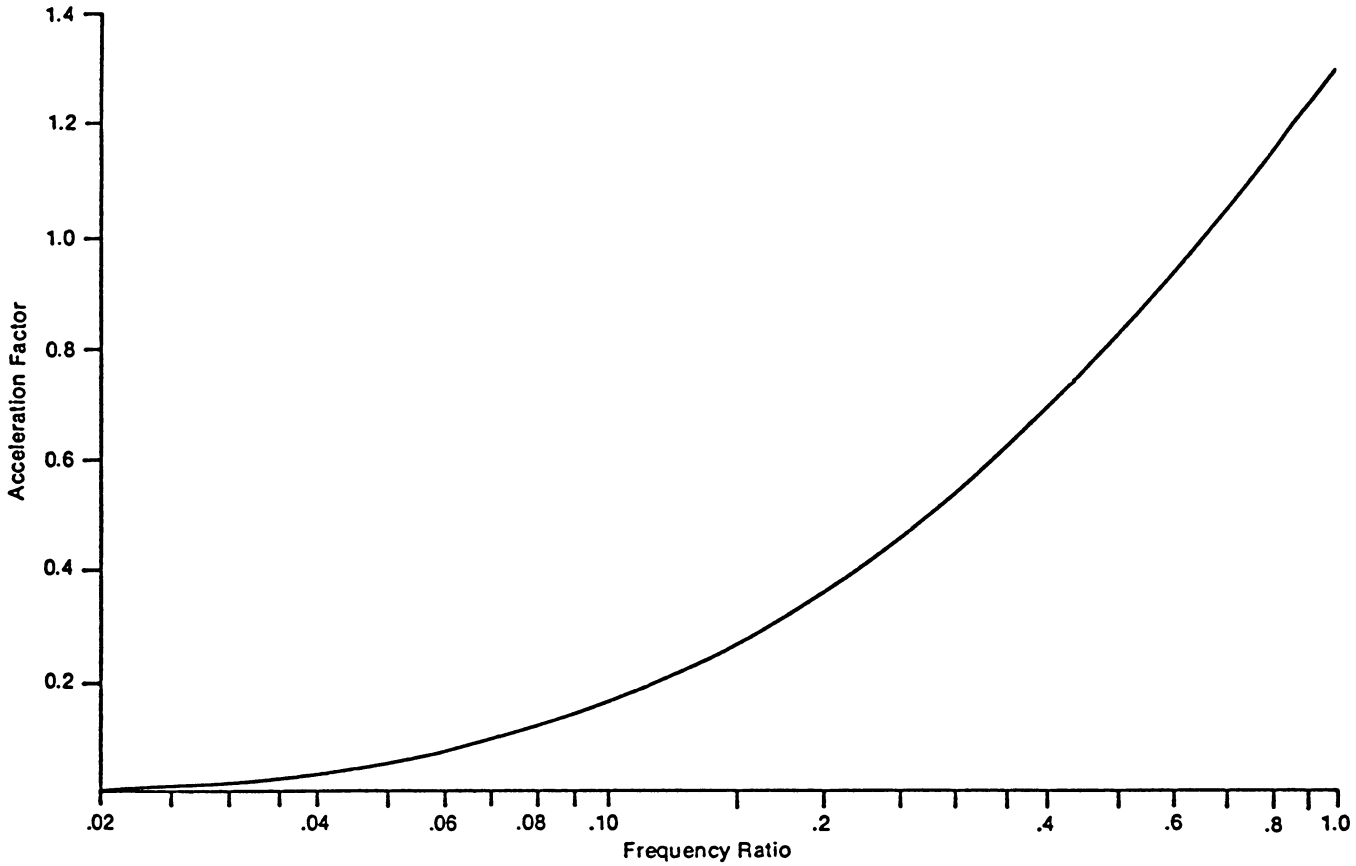


Figure 7. Shock Transmissibility

curves (a typical set of curves is shown in Figure 6) for the isolator you have selected. Locate the applied load value on the appropriate graph (45 degree Compression/Roll, Compression, Shear).

• The isolators that meet your requirements will have load/deflection curves that intersect the vertical coordinate for the minimum static deflection you have established. Calculate the resonant frequency to verify that it is less than one-fourth of the forcing frequency.

15. How about an example of a solution to severe service problems?

You can determine the amount of shock a unit will actually experience by calculating the Shock Response Frequency (F_S) of the isolation system. As in vibration calculations, the ratio of the F_S to the input shock pulse frequency is critical.

For example:

A 140 g, 3-msec half-sine pulse has a pulse frequency of 166 Hz (1,000 msecs divided by 6). An isolation system having a Shock Response Frequency of 20 Hz gives a ratio of 0.12. Reading from the graph (see Figure 7) indicates a shock transmissibility of 0.2, resulting in 28 g's response.

Because of the softening characteristic of cable isolators, the F_S is approximately 0.75 of the vibration natural frequency, and may be used to preselect an isolator. For instance, a system with a natural frequency of 20 Hz will have a F_S of 15 Hz. Thus, if the external vibration is intermittently interspersed with shocks (a truck traveling first on a smooth highway and then on rough terrain exemplifies this situation), the isolated equipment will experience less than 0.2 of vibration forces, and less than 0.3 of bounce forces.

Generally speaking, the isolation system should react progressively softer as the shocks increase in severity to limit the shock response to tolerable levels. Shock conditions of this kind are difficult to predict, but there are good guidelines for organizing isolation systems of this type:

• Restrain and damp the pitching motion of a tall unit (one whose height is more than twice its narrow width) with stabilizing mounts near its top. Arrange the load with the heaviest components nearest the base to minimize rocking.

• These stabilizers should provide lateral stiffness but should not limit vertical play. Too much lateral deflection at the top will change the characteristics of the entire system because the base mounts will then have to supply all the resistance to overturning moments.

• Locate the base mounts so that they balance the static load vertically. The isolators should be under or in line with the unit's vertical structural members. In the case of cabinets or consoles fabricated of sheet metal, it may be necessary to attach a supplementary base plate to bring the shock load into the rigid columns and avoid using the panel enclosures to distribute the shock.

• Spread the base mounts to provide the largest possible footprint for uniform support. Because shock may occur at an oblique angle, allow sufficient room to sway in all directions.

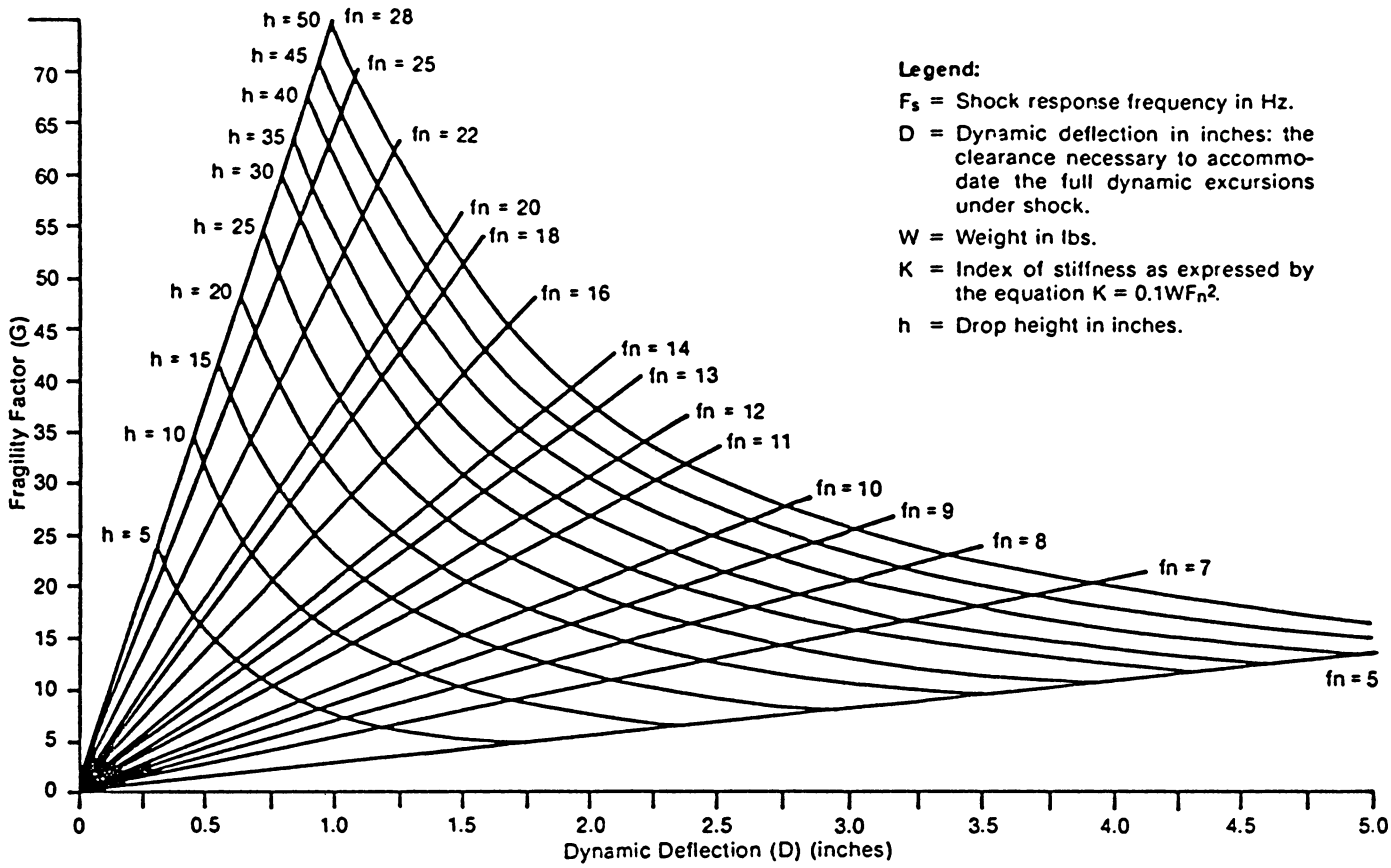


Figure 8. Nomograph

16. Should I select my own isolators or get engineering help?

You should be able to select the right isolator for most of the shock and vibration problems you are likely to encounter by turning to manufacturers' literature and selection guides. It may be easier than you think. Some isolators are forgiving, and rough estimates of shock and static loadings, frequencies, and amplitudes may be enough. In any case, manufacturers will confirm your choice by running tests and providing reports and comments.

To help you solve more difficult isolation problems, some manufacturers offer complete shock and vibration engineering service. They work with you cooperatively or shoulder the entire design responsibility.

The approach usually encompasses three steps. First, evaluate the problem. Second, create a prototype. Third, test it under conditions that simulate intended service as closely as possible.

17. Are there any unbiased selection guides?

Obviously, most manufacturer selection guides are based on extensive testing and experience in the field. Selection guides that include nomographs, tables, and other comparative information appear to be the most unbiased. However, you should always consider making your own comparative analysis of the isolators supplied by competing manufacturers.

18. How about a sample of a "Selection Aid"?

The nomograph presented in Figure 8 will help you determine deflection and response for a variety of environments. It covers load conditions, aircraft landings, shipping containers, and Navy tests. Knowing the severity of the applied impact, the designer can preselect the characteristics of the isolation system. From these, he can evaluate the resulting deflection and response in g's. In the same way, knowing the space available for deflection, he can find the necessary stiffness of the isolation system.

For the asking, Aeroflex Laboratories Inc. will supply a number of typical examples that walk you through solutions using this nomograph.

19. Where can I get help when I need it?

Obviously, help is as close as the nearest telephone. Most suppliers are staffed with experienced applications engineers, who will respond to your questions. Responses may be as simple as a single calculation, or as involved as a computer study—even the design and fabricating of prototype mounts if no existing design appears to fit the need.

20. What are the "stakes" when faced with selection of an isolation system for severe service?

The stakes are high. Often, protected equipment is crucial to the successful operation of hierarchy systems that are, in turn, crucial to safe, efficient performance.

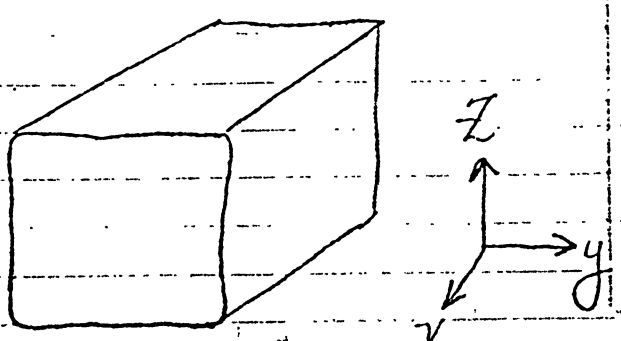
There, then, are our 20 most-often sought answers. Obviously, there are dozens more questions needing answers before the optimum isolator system becomes a reality. But we hope this information provides a starting point for your important decision-making process.

APPENDIX D:

BALL FACILITY REPORT

BLANK

m Page No.



TEST PROGRAM: DATE STARTED 18 DEC 86
PART NO. MIL 1553 SERIAL NO. SPERRY #2
TEST SPEC. NO. PROCEDURE NO.
PARAGRAPH NO. PARAGRAPH NO.
FIXTURE NO. MARTIN INCS BACK FIXTURE
SINE VIBRATION: FILTERED UNFILTERED
RANDOM VIB: EXTRA FILTERS REQUIRED
(INCLUDE DETAILS WITH EACH AXIS)

.5g SINE SWEEP 5-500 HZ

X-AXIS: SET UP FIXTURE ON LARGE SLIDE PLATE - MOUNT UNIT IN FIXTURE - MOUNT 3 RESPONSE ACCELS AND ONE (1) CONTROL ACCEL. - SET UP .5g SWEEP 2 OCT/MIN - 5-500 HZ - RUN 3 SWEEPS & PLOT DATA ACCELS LIVE - SET UP .9" ON 1.5g SINE SWEEP - RUN SWEEP & RECORD ACCELS ON TAPE & PLOT DATA 1 LIVE. SET UP CURVE AS FOLLOWS: 8-38 HZ @ 2.5g, 38-50 HZ @ .033" DA, 50-500 HZ @ 4.2g PEAK - 2 OCT/MIN - RELOAD ACCELS ON TAPE & DATA 1 LIVE. - COMPLETED SWEEP. SET UP SHOCK 20g 11 ms 1/2 SINE SHOCK 4 TIMES AT 20g 11ms - TAKE PLOTS OF RESPONSE ACCELS. SET UP 30g 11 ms - NO SHOCKS POSSIBLE @ 30/11 - DO 20g 11ms 3 TIMES PLOT FATAL RESPONSE. AFTER INSTALLING ADDITIONAL MOUNTS SET UP .5g SINE REL NOISE - DO 3 SWEEPS & PLOT DATA RESPONSE - LOAD RANDOM SPEC AS FOLLOWS

20-100 HZ @ .0003 G^2/HZ
100-300 HZ @ +4 dB/OCT
300-1000 HZ @ .04 G^2/HZ
1K-2K HZ @ -6 dB/OCT

RUN 10 MN RANDOM @ 7.3 GRMS PLOT CONTROLS & 3 DATA @ -6dB - ALSO PLOT CONTROLS & DMS @ -0 dB

1/19/86 - MOUNT UNIT & ACCELS - SET UP 10g 10ms SHOCK - OK - DO REPETITIVE PULSES - 3 EACH ONE SECOND APART - DO 10 EACH

To Page No. 8

m Page No. 4

SET UP 20g 10ms PULSE - OK
DO 5 PULSES @ 15g 10ms
DO 3 PULSES @ 20g 10ms
DO 10 PULSES @ 20g 10ms
DO 10 PULSES @ 20g 10ms 6 SECS APART

LOAD SINE .1g 5-2000Hz @ 2 OCT/MIN OPERATING
RUN SINGLE SWEEP TO ~800 Hz ABORTED
RAISE LEVEL TO .2g AND RUN SWEEP
RAISE LEVEL TO .4g AND SWEEP FROM 500 - 5Hz
@ 1 OCT/MIN
RAISE LEVEL TO .6g @ 1 OCT/MIN SWEEP
TO APPROX 25Hz - ABORT ON REQUEST

Y-AXIS: ROTATE FIXTURE 90° ON SIDE
PLATE MOUNT WITH ACCELS AS
FOLLOWS:
CHAN 1 - DRIVES BACK
CHAN 2 - PIONEER
CHAN 3 - POWER SUPPLY
LOAD .8g SINE 5-500Hz @ 2 OCT/MIN
SWEEP (3) TIMES & GET DATA PLOT
ATTEMP 2 RUNS STARTING @ 5Hz
NO GO - TOO MUCH SLIDING IN
SLIDE PLATE - START @ 500Hz
SWEEP DOWN TO 5Hz
GOTO .5g 5-500Hz @ 2 OCT/MIN
MAKE (3) SWEEPS TO PLOT 3
DATA ACCELS

SET UP 20G 10MS ½ SINE STOCK TO
4 EACH 10G 10MS WITH DATA
3 EACH 15G 10MS WITH 1 DATA } D.TH 1
3 EACH 20G 10MS WITH 1 DATA }

To Page No.

Witnessed & Understood by me,	Date	Invented by	Date
		Recorded by	

m Page No. 8 21 DEC-86

Y AXIS: SET UP RANDOM AS BEFORE ON PAGE 7 — RUN -6dB DOWN FOR APPROX. 5 MIN TO RECORD DATA ACCELS. INSTALL SLICK PINS & REPEAT -6dB RUN

Z AXIS: ROTATE EXCITER VERTICAL - MOUNT SM HEAD EXPANDER & MOUNT FIXTURE INSTALL UNIT AND CONNECT ACCELS DO .5g SWEEP 5-500HZ AS BEFORE - DO THREE SWEEPS - GET DATA ACCELS INDIVIDUALLY DURING RUNS MOVE CONTROL TO FIXTURE BASE DO Z DATA RUNS WITH NEW CONTROL POINT

SLICK - SET UP 20g 10ms SLICK - OK DO 4 PULSES 15g 10ms TO GET DATA & INPUT DO 2 PULSES @ 20G 10 ms

RANDOM - SET UP RANDOM SPEC PER PREVIOUS TEST — RUN @ -6dB DOWN AND TAKE DATA - STEP UP TO FULL LEVEL AND TAKE DATA

TEST COMPLETE

tnessed & Understood by me,	Date	Invented by	Date
		Recorded by CJZ SCHLING	12/22/86

BLANK

BLANK

BLANK